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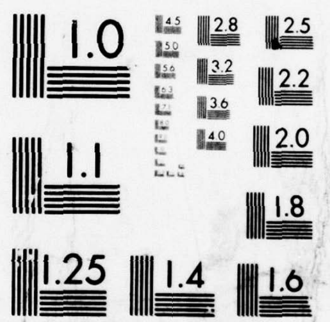

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DEMONSTRATION PROGRAM FOR A FLEXIBLE DUCT VALVE FOR RAMJET ENGINE FUEL CONTROLS

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A demonstration program was conducted to develop a flex duct-type metering valve for use in the fuel control of ramjet engines for Navy missiles. This type valve has inherent advantages of simplicity, low cost, and high reliability that make it particularly attractive for this application. This type valve functions best when the leakage flow from the metering gap is bypassed back upstream of the fuel pump.

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Evaluation of the flex-duct valve disclosed that, for fuel systems requiring high pressure and high flow rates, the spring rate of the flex duct will of necessity be quite high. The high spring rate of this type of valve therefore makes it impractical for applications requiring a high valve response rate and low actuation force levels. The requirement for a high valve response rate in the ramjet engine fuel control pointed toward the application of a flexure-type throttle valve. This type of metering valve offers greater simplicity, lower cost, high reliability and meets the high system response rate requirements of the ramjet engine at low actuation force levels. The flexible-type throttle valve is therefore presented in the final analysis of this demonstration as the type of metering valve that can best achieve the fuel metering requirements of a ramjet engine fuel control.

EXECUTIVE SUMMARY

PROGRAM OBJECTIVE

This program was designed to develop a flex-duct type metering valve to be used in the fuel controls of ramjet engines for missile applications. The flex-duct metering valve was selected for development because of simplicity, reliability, ease of manufacture, and low cost.

DESIGN ANALYSIS

Three types of duct valve were evaluated in the initial design selection phase of the program. These were the jet-pipe, flex-duct, and the flexure-type throttle valves. The flex-duct and flexure-type throttle valves showed the least development risk for achieving the design objectives. Two designs of each valve were fabricated and tested. The most critical factors in meeting the design objectives were:

a. High fuel pressure (1500 psi maximum)

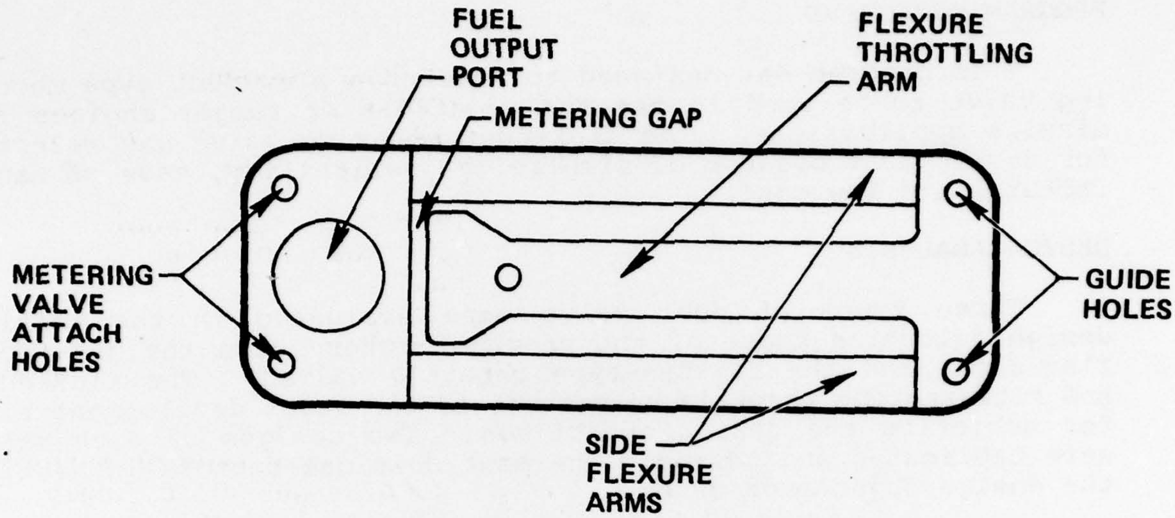
b. Valve response rate required to obtain the required fuel scheduling accuracy at a low actuator force level.

c. Hysteresis

d. Turndown ratio (ratio of maximum to minimum flow)

The flex-duct valve, when designed to contain the high-fuel pressure, required a high actuation force to achieve the required fuel scheduling accuracy. A 200 psi maximum fuel pressure is practical for this type metering valve. Above this pressure, it becomes necessary to increase the duct wall thickness, thus making the flexure spring rate high and requiring a high actuation force for metering fuel.

Attempts to overcome this problem resulted in designing the spring separate from the duct flow channel. This increased the complexity of the valve with very little reduction in size or weight, and no significant improvement in performance; however, it was possible to reduce the actuation force level. Separation of the flexure spring from the flow channel led to the design and test of the flexure-type throttle valve. Initial testing of a modified flex-duct valve proved that this concept satisfied all the design objectives; however, the valve was larger and heavier than necessary. Therefore, the small compact flexure-type throttle valve ("W"-shaped valve) was designed, fabricated, and tested. A schematic of the valve is shown below:



Flexure-Type Throttle Valve

This type of metering valve proved to be highly satisfactory and was designed for use in the ramjet engine fuel controls for the air-to-surface and air-to-air missiles used at the Naval Weapons Center at China Lake, California.

TEST RESULTS

The "W"-shaped metering valve met all design objectives for the ramjet engine fuel control for the air-to-surface missile application. The flex-duct metering valve is very successfully being used in the fuel control of the AiResearch GTCP85-180 APU gas turbine engine. It is also being used in the inlet guide vane (IGV) control of the AiResearch GTCP36-200 APU gas turbine engine. In both of these applications, the pressures of the working fluid are well below 200 psi.

A turndown ratio of 4.8:1 (ratio of maximum to minimum flow) was achieved with a "W"-shaped throttle valve tested in this program which is more than sufficient for the application in which the valve is now being used. The controlling design characteristics which affect the turndown ratio are the angle that the metering gap makes with the flow path in the receiver port, and the minimum metering gap. The angle of the metering gap used in the throttle valve in this program was 90 degrees (normal to the flow path). The metering gap required to achieve a turndown ratio of 4.8:1 was 0.0035 inch.

Sufficient testing of the metering gap cut at an angle was conducted to show that turndown ratios above 20:1 can be achieved with this type metering valve.

CONCLUSIONS

The flexure-type throttle valve and the flex-duct valve will meter fuel for a wide range of applications including a ramjet engine fuel control. The flexure-type throttle valve can be designed to withstand fuel pressures well above 1500 psi with good fuel scheduling accuracy at low actuation force levels. The flex-duct valve is a very effective metering valve for applications where fuel pressures are below 200 psi. The most difficult task in designing this type of valve is that of hysteresis control. Hysteresis in this design was controlled sufficiently to achieve the desired fuel metering performance; however, attention to the design of the valve actuator can result in effecting even greater reduction of hysteresis.

Good control of the turndown ratio with this type of metering valve can be achieved by cutting the metering gap on an angle and by controlling the minimum gap for minimum fuel flow.

The flexure-type throttle valve was demonstrated to be very simple to design, easy to manufacture, highly reliable, and possesses a good potential for low cost as compared to other types of metering valves for ramjet engine fuel control applications.

PREFACE

This is the final report of a demonstration program for a flexible duct valve to be used on a fuel control for a ramjet engine. The program was conducted by AiResearch Manufacturing Company of Arizona, a Division of The Garrett Corporation, for the Aircraft Crew System Technology Directorate, Naval Air Development Center (NADC), Warminster, Pennsylvania, and in connection and complimentary effort with the Naval Weapon Center, China Lake, California. The work covered by this report was performed under Navy Contract N62269-77-C-0352.

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FIGURES

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TABLES

1. INTRODUCTION AND BACKGROUND

1.1 INTRODUCTION

The scope of work to be accomplished in this demonstration program encompassed the design, fabrication, and test of a flexible duct-type metering valve for a ramjet engine fuel control.

1.2 BACKGROUND

Based on production experience of similar type applications, the flexure-type metering valves have the following advantages:

a. Permit low manufacture cost.

b. Provide high reliability.

c. Are simple and minimize the use of sliding or contacting parts.

d. Have flow efficiencies approaching the conventional spool and sleeve metering valve.

e. Are not susceptible to clogging from contamination.

Alternative flexible-type fuel-metering valves were studied in the process of selection of a valve for design and test. The three types of valves considered were the jet pipe, the flex duct, and the flexure-type throttle valve. The jet pipe and flex duct valves shown in Figures 1(a) and 1(b) have moveable flow channels which discharge fuel into a recovery port. Metering is achieved by mechanically displacing the flow channel normal to the axis of flow in proportion to the flow demand of the engine. This action may either restrict the flow to the recovery port or bypass a portion back to the fuel pump. The flexure-type throttle valve shown in Figure 1(c) meters fuel from the fuel chamber into the discharge port by mechanically displacing the metering flexure across the face of the port in proportion to engine fuel flow requirements.

The selection parameters used to determine which valve should be carried into development and test were simplicity, reliability, performance, and cost. The jet pipe metering valve was considered as an alternative in the early concept selection stage of this program; however, it was discarded because of unsatisfactory experience with friction forces, hysteresis, and thermal growth of the flow channel.

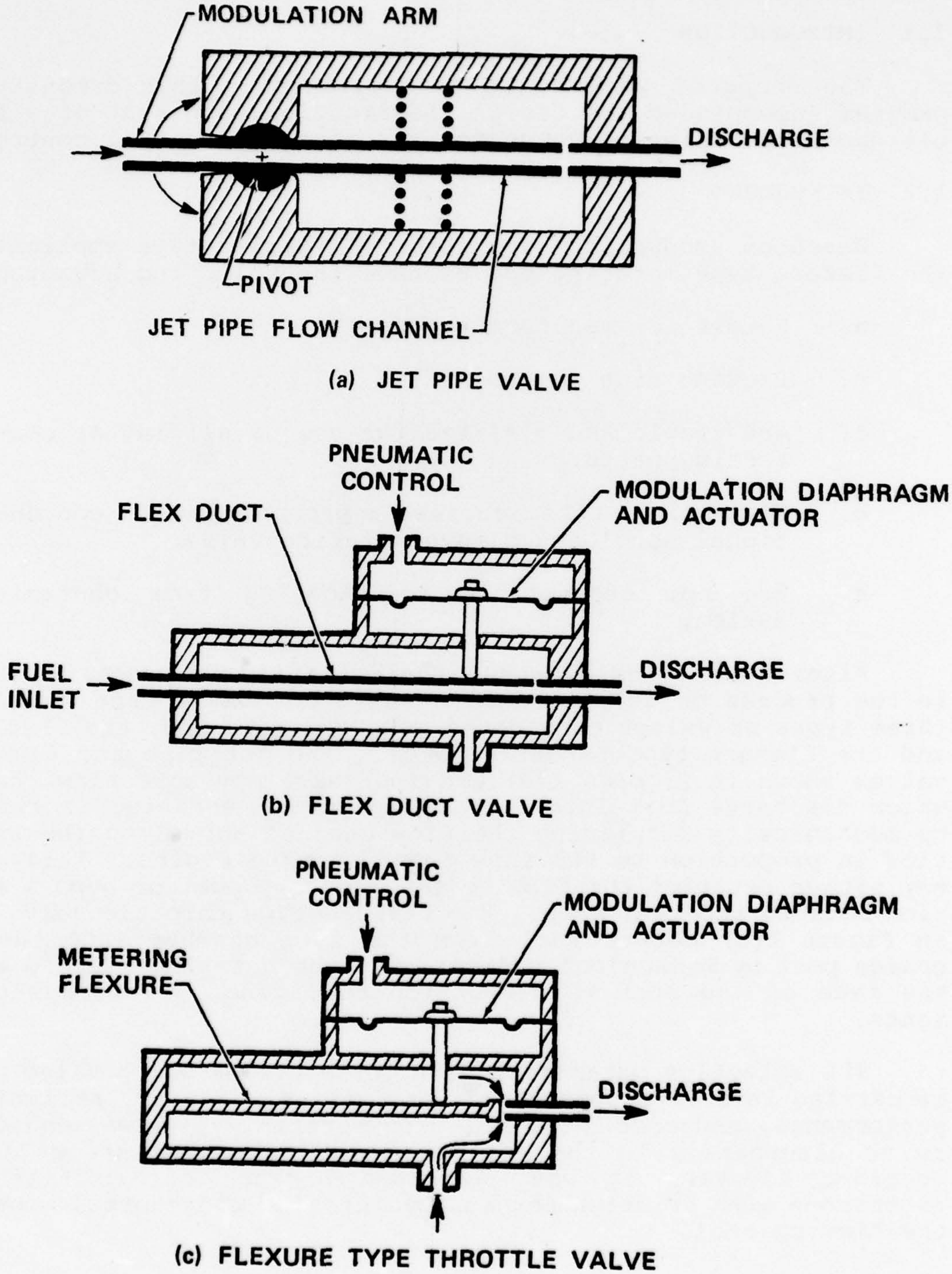


Figure 1. Metering valves.

2. TECHNICAL ANALYSIS

2.1 PERFORMANCE OBJECTIVES

Typical ramjet engine performance requirements, as given in Table 1, were selected as a baseline for the design of the fuel metering valves evaluated in this program. The working fluid for tests conducted in this program was JP-4 calibrating fluid per MIL-L-7024.

TABLE 1. FUEL METERING VALVE PERFORMANCE PARAMETERS.

Turndown ratio $\left(\dfrac{\text{Maximum Flow}}{\text{Minimum Flow}}\right)$	3.2:1
Maximum flow rate	2.706 lb_m per sec
Minimum flow rate	0.842 lb_m per sec
Fuel pressure range	200 to 1000 psi (inlet pressure)
	100 to 400 psi (output pressure)
Proof pressure	1500 psi (inlet pressure)
Allowable pressure drop:	
at 200 psi	~100 psi
at 500 psi	~125 psi
at 1000 psi	~155 psi
Ambient temperature range	minus 40F to plus 160F

2.2 DESIGN ANALYSIS

2.2.1 <u>Valve Configuration</u>

The two basic configurations of fuel metering valves evaluated for the ramjet engine fuel control application were the flexible-duct valve and the flexible-type throttle valve [see Figures 2(a) and 2(b)].

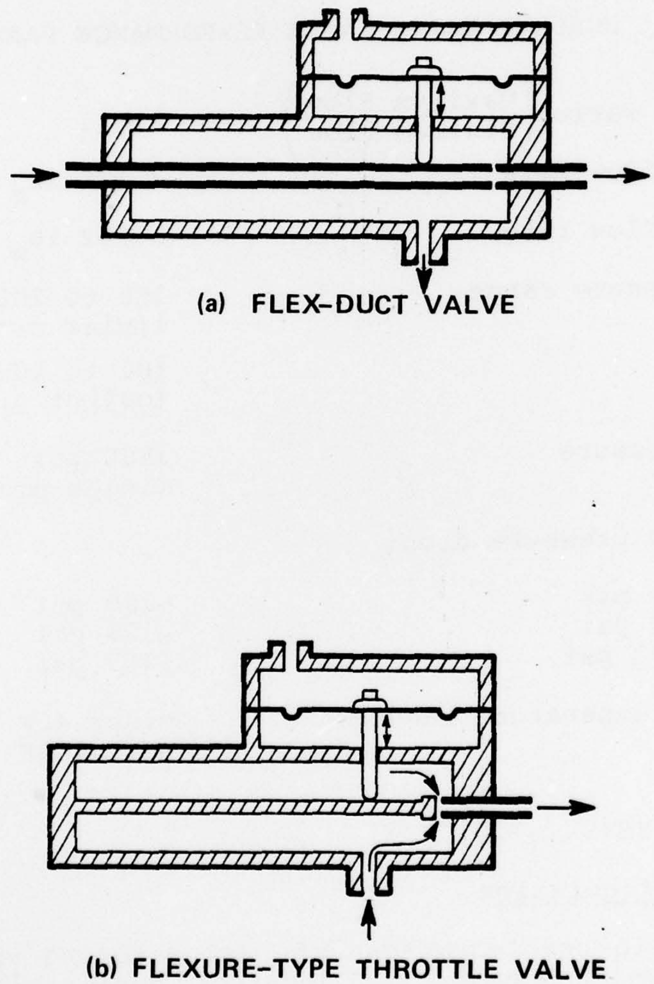


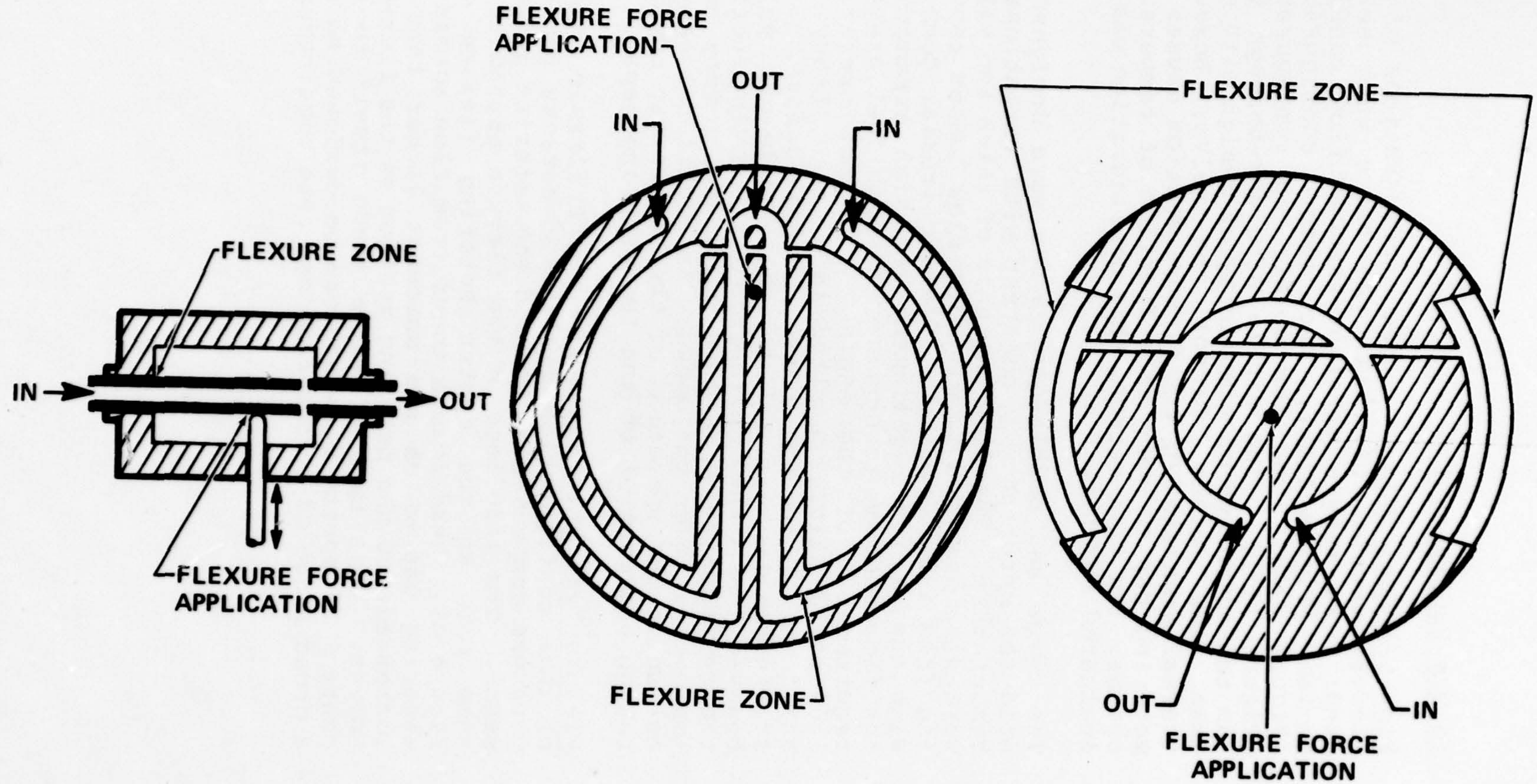
Figure 2. Metering valve configurations.

2.2.1.1 Flexure Configuration

a. Flexible-Duct Valve - Three configurations of the flexure section of this type valve were evaluated: the single straight-through flow configuration [Figure 3(a)], the omega configuration [Figure 3(b)], and the double-gap configuration [Figure 3(c)]. The single straight-through flow has the advantage of simplicity of design with minimum flow losses through the valve; however, thermal expansion of the flexure section causes the metering gap to vary as a function of temperature change. For this reason, this configuration was not evaluated further.

The omega and double-gap valves were designed to hold the metering gap constant with variations in temperature. The flow channels of these two valves were also designed to minimize flow losses resulting from flow reversal. These criteria, combined with the design requirement for high strength in the flow section to support the high fuel pressure requirements of the engine (1200 psi operating, 1500 psi static), resulted in a large, heavy, and stiff valve in the omega configurations. The stiffness was reduced in the double-gap valve; however, due to the high inertia of the moving section, the response rate of this valve was a very small improvement over the omega. An additional complication in the operation of the double-gap configuration was control of the two metering gaps.

b. Flexure-Type Throttle Valve - The flexure section of this configuration has the advantages for temperature compensation to hold the metering gap constant. The side legs of the flexure expand at the same rate as the center metering flexure (see Figure 4). Head losses occur from flow across the metering gap which are somewhat larger than the losses across the metering section of the flex-duct valves. This type of valve lends itself to easy control of spring rate and can be designed to meet a broad range of system response rate requirements.



(a) SINGLE STRAIGHT-THROUGH FLOW

(b) OMEGA

(c) DOUBLE GAP

Figure 3. Flexible-duct metering valves.

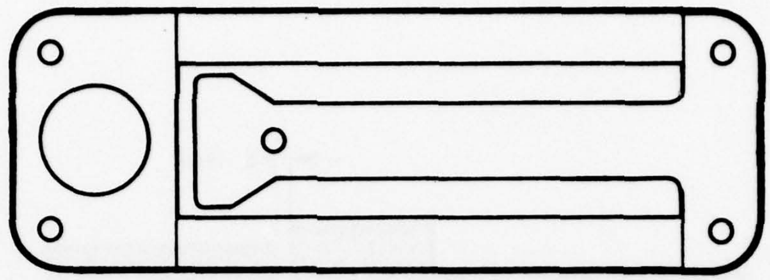


Figure 4. Flexure-type throttle valve.

2.2.1.2 Gap Configurations - The metering gap of either the flex-duct or the flexible-type throttle valve may be straight cut or angle cut as depicted in Figures 5(a) and (b). Factors to be considered in the gap selection are:

- o Requirement for fuel shutoff
- o Fuel metering rate as a function of flexure stroke
- o Turndown ratio required (maximum to minimum fuel flow)

a. Straight Cut Gap - The straight cut gap has a minimum flow rate which is dictated by the minimum gap achievable to obtain the required turndown ratio. If the gap is too small, the flexure will strike the fixed port before achieving minimum flow in the case of a flex-duct valve and before achieving maximum flow in the throttle valve. If the gap is too large, the bypass ratio becomes excessive in the flex-duct valve, and minimum flow becomes difficult to achieve in the throttle valve. The straight cut gap is the simplest to fabricate and calibrate if the desired turndown ratio can be achieved.

b. Angle Cut Gap - The angle cut gap is more difficult to fabricate and to calibrate; however, fuel shutoff can be achieved with this type of gap if required. One hundred percent shutoff however is difficult to achieve without the application of a strong spring force. This strong shutoff force then becomes a complicating factor to overcome with

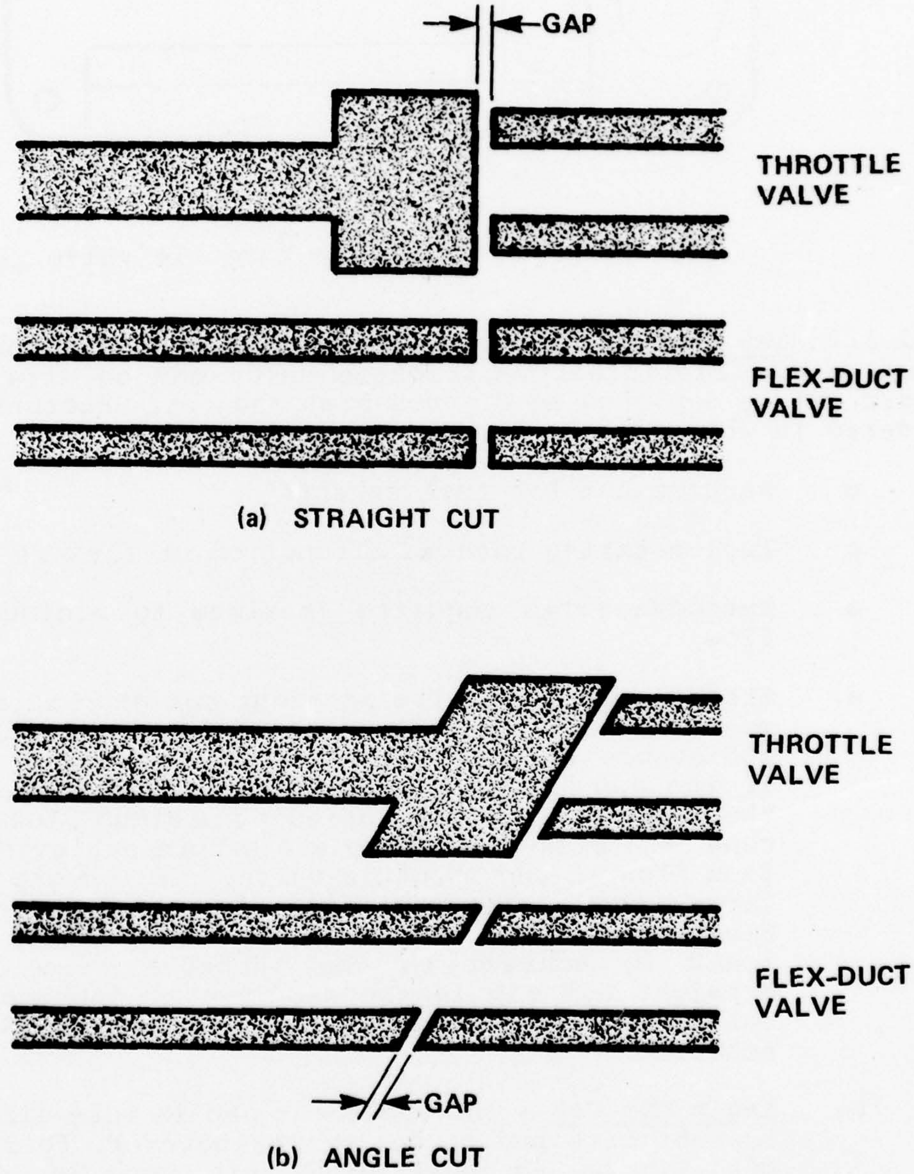


Figure 5. Gap configurations.

light valve modulation forces. The degree of movements with an angle cut gap to achieve a high turndown ratio is less than the straight cut. The straight cut gap, if adaptable to the metering requirements of an engine, is simplest and most desirable; however, if the required turndown ratio is too high to achieve with a straight cut gap, the angle cut may be used.

2.2.2 Flow Calculations

The flexible-duct valve has a rectangular flow channel which is cantilevered and flexible because of its laminated construction. The flow channel is separated from the receiver by a gap. The flexible-duct is actuated to meter fuel by a mechanical push rod. The leakage flow past the clearance gap (bypass flow) is returned to the supply source. A source of inefficiency is leakage through the gap.

This leakage flow is proportional to the leakage area shown in Figure 6, and the square root of the pressure differential across the gap according to the following equation:

$$W_L = C_L A_L \sqrt{2g_c \rho (P_2 - P_V)} \qquad (1)$$

where:

W_L = Leakage flow (lb/sec)

C_L = Discharge coefficient of the gap

A_L = Area of the gap (in.^{2})

g_c = Gravitational constant (lb_m ft/lb_f sec^2)

ρ = Fuel density (lb_m/in.^{3})

P_2 = Static pressure of fuel in duct at gap (lb_f/in.^{2})

P_V = Vent pressure outside the duct (lb_f/in.^{2})

The flex-duct flow equation for incompressible flow is expressed by Equation (2) below (see Figure 6):

$$W_1 = C_D A \sqrt{2g \rho (P_1 - P_2)} \qquad (2)$$

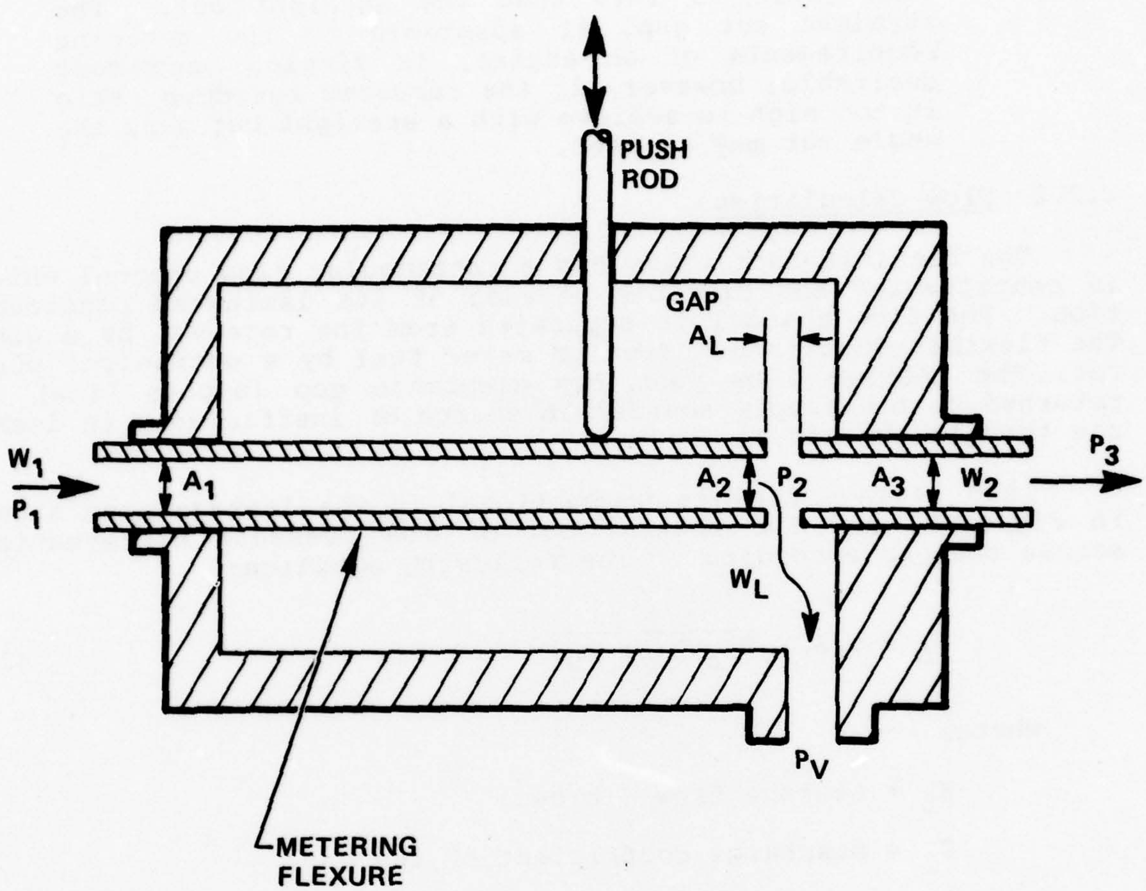


Figure 6. Schematic of flex-duct valve.

where:

W_1 = Duct flow (lb_m/sec)

C_D = Duct discharge coefficient

A_2 = Duct metering area (max $A_2 = h \times b$) (in.^{2})

P_1 = Upstream static pressure (lb_f/in.^{2})

h = Duct height of flow channel (in.)

b = Duct width of flow channel (in.)

The downstream load flow is determined by Equation (3) (see Figure 6).

$$W_2 = \frac{A_2 \sqrt{2g_c \rho (P_2 - P_3)}}{E (1 + A_2/A_3)} \quad (3)$$

where:

W_2 = Discharge flow (lb_m/sec)

A_3 = Outlet area (in.^{2})

P_3 = Downstream discharge pressure (lb_f/in.^{2})

E = Downstream flow efficiency

Flow continuity is satisfied by Equation (4):

$$W_1 = W_2 + W_L \quad (4)$$

Using the flow and pressure requirements of the engine, the flexure channel cross-sectional area of the flex-duct valve (Figure 6) may be calculated from Equations (1), (2), and (3). The fuel pump characteristics sufficient to support the flow and pressure requirements of the metering valve are assumed. Additional flow characteristics which must be iterated to size the flex-duct metering valve are:

a. Turndown ratio--maximum to minimum flow

b. Flow channel aspect ratio--height-to-width ratio (h/b)

c. Friction and duct losses (pressure and flow)

d. Metering gap effects

   1. Thermal effects

   2. Angle of gap with respect to flow path

e. Recovery channel diffuser effect

f. Flow forces

   1. Momentum

   2. Bernoulli

The flow calculations for a flexure-type throttle valve shown in Figure 7 are simpler than for the flex-duct valve shown in Figure 6 because there is no leakage or bypass flow to deal with.

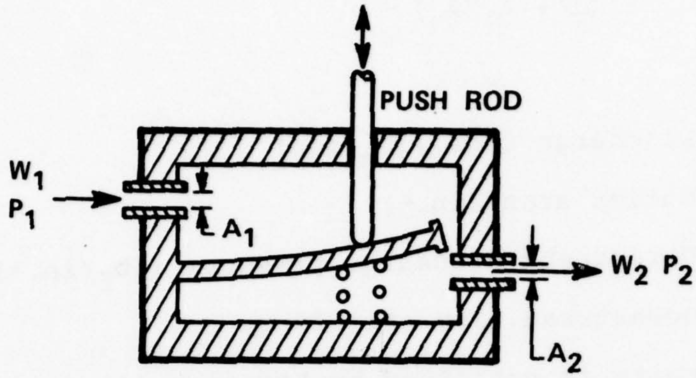


Figure 7. Flexure-type throttle valve.

2.2.2.1 Turndown Ratio - Turndown ratio is identified as the maximum fuel flow rate required, divided by the minimum fuel flow rate. The turndown ratio is established by the engine fuel flow requirements. The turndown ratio of a bypass flexible-duct type metering valve must be calculated directly from maximum and minimum fuel flow; however, the turndown ratio for a throttle valve may be calculated for a constant differential pressure across the metering section by dividing the maximum flow area by the maximum leakage flow area. The design criterion for metering valves evaluated in this program was to meter fuel at a maximum flow rate of 2.7 lb_m/sec with a turndown ratio of 3.2:1.

2.2.2.2 Flow Channel Aspect Ratio - Aspect ratio is defined as the ratio of channel height (h) to the channel base (b). The optimum aspect ratio may be selected as a function of flexure bending stress, metering valve spring rate, and flow friction. Experience with fluid flow in fluidic circuits has indicated that friction forces for small ducts become excessive at aspect ratios below 0.40 (h/b = 0.40). As shown in Paragraph 2.2.3, Mechanical Properties, the flexure section spring rate increases as a function of the third power of the flow channel height. Therefore, an aspect ratio of 0.40 was used in sizing the flow channel of valves evaluated in this program to maintain the flex-duct spring rate as low as possible. Paragraph 2.2.3 also shows that the bending stress is directly related to the flexure channel height (h). This relationship also dictates that channel height be held as small as possible to achieve an ultimate design stress level that will provide an adequate fatigue life for the required application.

2.2.2.3 Friction and Duct Losses - The head loss or the drop of pressure in a length of duct results from friction forces between the flow of fluid along the channel walls, and any sudden change in direction of the fluid stream flow. The duct pressure drop resulting from friction losses is computed using the Moody chart (Figure 8). Other losses not directly associated with the friction losses are called minor losses. These losses, such as sudden expansion or contractions, 90- or 180-degree bends, may be calculated using empirical test data. The overall pressure drop for a valve is a summation of each incremental pressure drop.

a. Duct Losses - Channel friction losses may be obtained by the use of the Moody chart, Figure 8. Pressure loss in a channel may be determined first by calculating Reynolds Number.

$$R = \frac{VD}{\nu}$$

where:

V = Fluid stream velocity = Q/ A (ft/sec)

Q = Flow (ft^3/sec)

A = Area of channel cross section ($in.^2$)

ρ = Density ($lb_m/in.^3$)

D = Hydraulic Diameter, = 4 A/P (in.)

P = Perimeter of channel (in.)

ν = Kinematic viscosity (ft^2/sec)

The relative roughness number (ϵ/D) is needed to enter the Moody chart.

where:

ϵ = Roughness factor given on chart corresponding to particular materials

D = Hydraulic Diameter = 4 A/P

The point where the relative roughness number intersects the Reynolds number defines the friction factor.

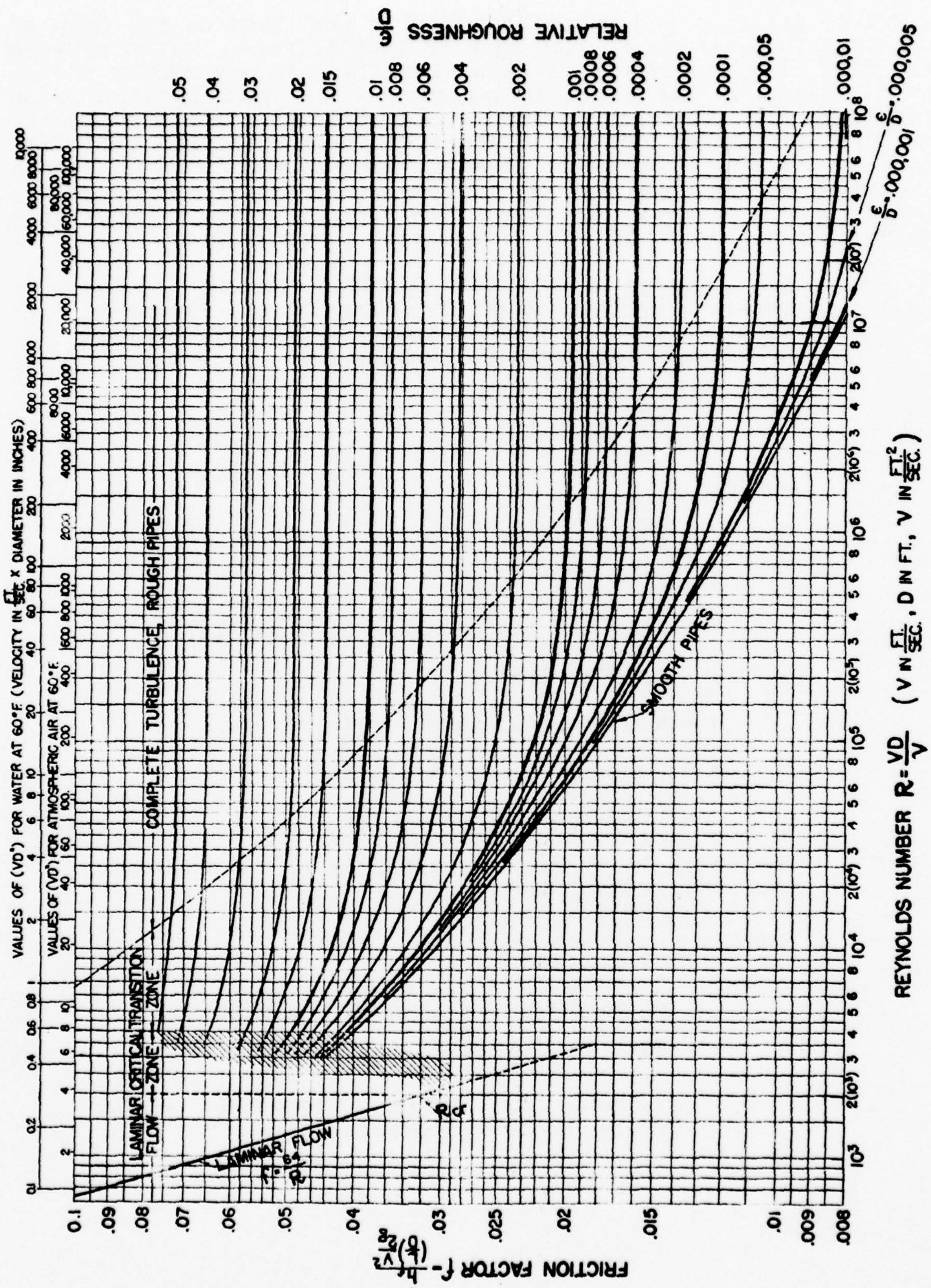


Figure 8. Moody chart.

Head loss due to channel losses may be determined from the frictional factor number.

$$h_f = f \frac{LV^2}{D2g}$$

f = Frictional factor

L = Length of channel (in.)

h_f = Head loss (in.)

The head loss expressed in terms of inches of water may then be converted to a pressure drop (psid) to represent the friction flow pressure drop in the channel.

2.2.2.4 Flow Forces - In a metering valve, flow is separated into two forces, momentum and Bernoulli forces. Momentum forces address those forces which result from changes in direction of the fluid stream velocity while Bernoulli forces are associated with change in pressure due to changes in the magnitude of the fluid stream velocity.

a. Momentum Forces - The equation for momentum forces for both the flexure type throttle and the flex-duct valve is:

$$F_f = C_d A \sqrt{2g_c \rho \, \Delta P} \quad \Delta V \sin\theta \quad (\text{lb x ft/sec}^2)$$

and is derived from the equations:

$$F_f = \dot{m} \quad V \sin\theta$$

$$\dot{m} = Q\rho, \text{ and}$$

$$Q = C_d A \sqrt{\frac{2g_c}{\rho} \Delta P}$$

where:

$\dot{m}$ = Fluid mass flow rate (lb_m/sec)

Q = Volume flow rate (ft^3/sec)

ρ = Fluid density ($lb_m/in.^3$)

g_c = Gravitational constant $\left(\frac{lb_m\ ft}{lb_f\ sec^2}\right)$

ΔV = Change in velocity for a given direction (ft/sec)

ΔP = Change in pressure across the metering section ($lb_f/in.^2$)

C_d = Discharge coefficient of the flow area

A = Flow area ($in.^2$)

θ = The angle of the incoming fluid relative to the horizontal flow axis of the receiver port (see Figure 9)

The horizontal flow force component thus is:

$$F_f = C_d A \sqrt{2g_c \rho\, \Delta P}\ \Delta V \cos\theta \quad (lb_m \times ft/sec^2)$$

and the vertical flow force (in the axis of the flexure valve actuation force) is:

$$F_f = C_d A \sqrt{2g_c \rho\, \Delta P}\ \Delta V \sin\theta \quad (lb_m \times ft/sec^2)$$

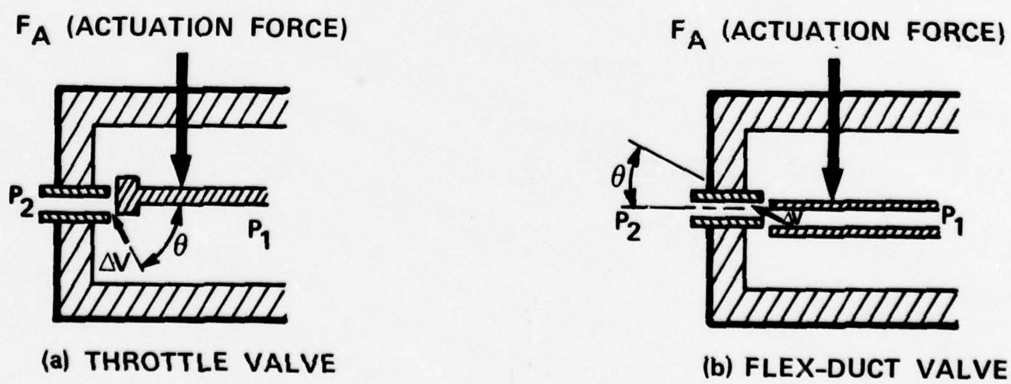


Figure 9. Flow force vectors.

The change in velocity may be calculated from the incompressible flow equation:

$$W = AV = \rho C_d A \sqrt{\frac{2g_c}{\rho} \Delta P}$$

$$V = C_d \sqrt{\frac{2g_c}{\rho} \Delta P}$$

$$\Delta V = V$$

where:

$$\Delta P = P_1 - P_2 \text{ ie}$$

$$\Delta V = C_d \sqrt{\frac{2g_c}{\rho} (P_1 - P_2)}$$

The flow force influencing the actuation force from below then becomes:

$$F_f = C_d A\ 2g_c\ \Delta P \sin \theta$$

Experience indicates that this flow force is most significant at small values of θ (very near the closed position for the throttle valve). In this region, θ may be calculated by:

$$\theta = \text{arc tan} \frac{\text{gap (in.)}}{\text{valve opening (in.)}}$$

The flow forces from leakage on the sides offset each other giving a zero net side flow force. The flow force from leakage past the gap at the top is at an angle of θ = 180 degrees. Since sin 180 degrees = 0, the flow force acting on the metering edge is zero.

The flow coefficient (C_d) is best obtained by force measurement from laboratory testing.

b. <u>Bernoulli Flow Forces</u> - Bernoulli forces occur as a result of a change in the magnitude of the fluid stream velocity due to pressure changes. The Bernoulli equation is:

$$\frac{P_1}{\rho} + \frac{V_1^2}{2g_c} = \frac{P_2}{\rho} + \frac{V_2^2}{2g_c}$$

where:

ρ = Fluid density $\left(\frac{lb_m}{in.^3}\right)$

g_c = Gravitational constant $\left(\frac{lb_m}{lb_f}\ \frac{in.}{sec^2}\right)$

P_1 = Upstream pressure (psi)

V_1 = Upstream velocity (ft/sec)

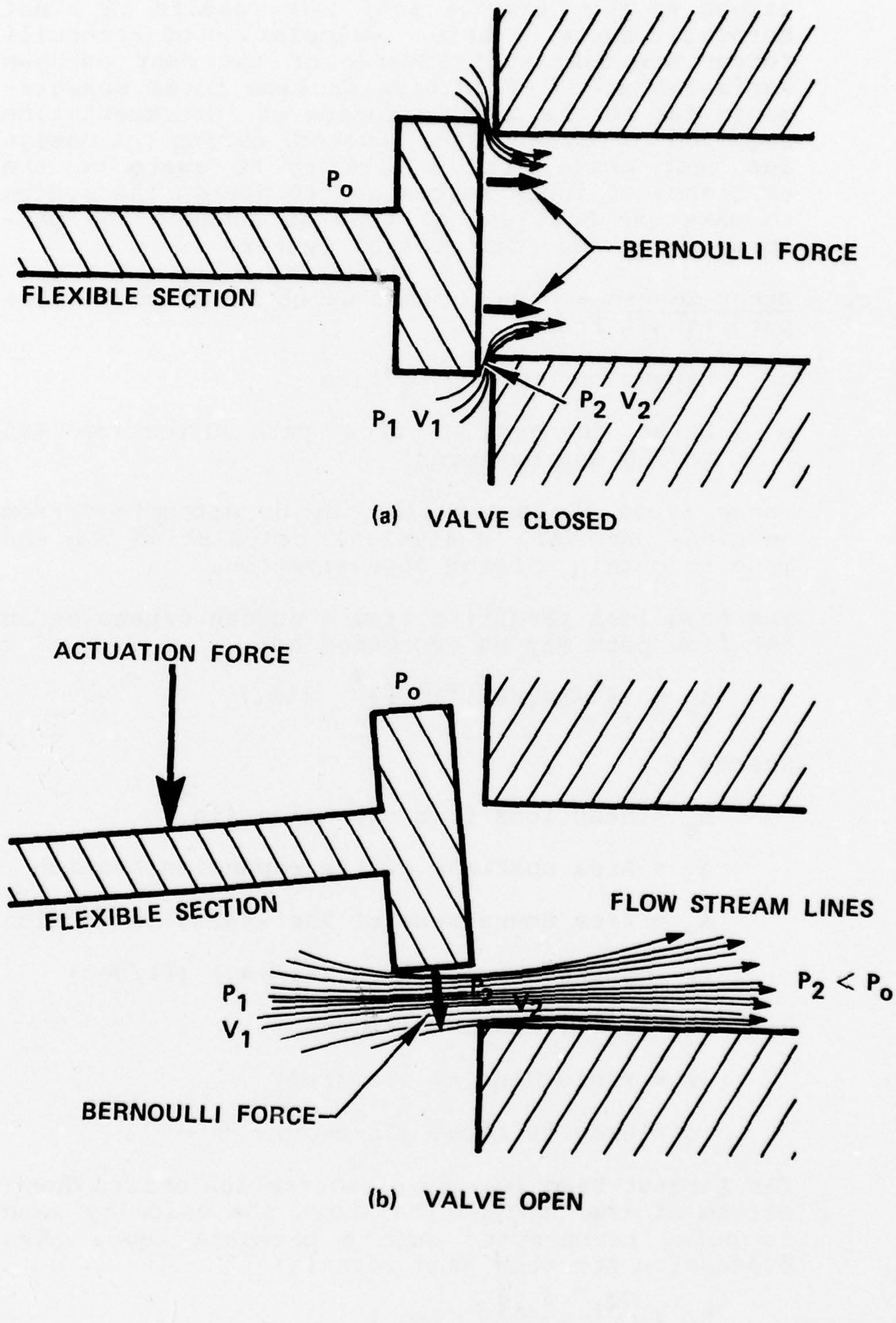
P_2 = Downstream pressure (psi)

V_2 = Downstream velocity (ft/sec)

Forces due to the Bernoulli force are present in two areas: (1) the gap area and (2) flow along the flexible metering arm.

A decrease in the gap width results in an increase in flow velocity; however, since the valve is rigidly attached, this force can be neglected [see Figure 10(a)].

As shown in Figure 10(b), the metering edge of the throttle valve (when closing) has a venturi effect on the flow stream. A Bernoulli force is created on the flexure, which is a result of the increase in flow velocity at the metering edge of the throttle valve. This force, added to the actuation force, causes a tendency for an overshoot of the commanded metered position. Fuel flow instability may result from the lag between the measured fuel flow and the commanded flow because of the Bernoulli force effect on the feedback loop of the control. Fuel flow instability can be prevented by designing the metering valve spring with a high spring rate. Also, the shape and contour (surface roughness) of the flexure, located at the metering edge the valve, may be designed to prevent the creation of a Bernoulli force.



(a) VALVE CLOSED

(b) VALVE OPEN

Figure 10. Bernoulli forces diagrams.

With a rough surface, the fluid stream will not attach to the flexible arm; this results in a net Bernoulli force of zero. Calculation of Bernoulli forces are difficult because of the many unknown variables and difficulties in flow force measurements due to the sophistication of instrumentation required to obtain data; however, during the design and test phase, it is well to be aware of the existence of these forces and to design the system to make the best use of them to enhance the performance of the fuel control system.

c. Other Losses - Other losses which occur in the flow path result from:

- Expansion and contraction
- Large changes in flow path direction (90 to 180 degree turns)

These types of losses also may be determined from testing; however, analytical calculation may be made to obtain a first approximation.

The head loss resulting from a sudden expansion in the flow path may be expressed as:

$$h_e = [1-(A_1/A_2)^2]^2 \frac{V_1{}^2}{2g} \quad \text{(in.)}$$

where:

h_e = Head loss from expansion (in.)

A_1 = Area upstream of the expansion section

A_2 = Area downstream of the expansion section

V_1 = Upstream velocity = $(W_1/\rho A_1)$ (ft/sec)

W_1 = Upstream flow (lb_m/sec)

ρ = Fluid density (lb_m/ft^3)

g = Gravity force (ft/sec^2)

The largest head loss in a contraction occurs downstream of the contraction where the velocity head is being reconverted into a pressure head. The expression for this head loss is:

$$h_c = \frac{(V_0{}^2 - V_2^2)}{2g} \quad \text{(in.)}$$

where:

V_0 = The stream velocity immediately downstream of the contraction

V_2 = The downstream velocity

A continuity equation expressing the impact of the coefficient of contraction is $V_0 \, C_c \, A_2 = V A$. From this relationship the head loss for contraction may be rewritten as:

$$h_c = \left(\frac{1}{C_c} - 1\right)^2 \frac{V_2^{\,2}}{2g} \text{ (in.)}$$

C_c = Coefficient of contraction expressed as a ratio of the downstream to upstream area A_2/A_1 (see Table 2 below).

TABLE 2. COEFFICIENT OF CONTRACTION OF FLOW IN A CHANNEL

A_2/A_1	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
C_c	0.624	0.632	0.643	0.659	0.681	0.712	0.759	0.813	0.892	1.0

The equation for the head loss due to smooth changes in the flow path direction (90 to 180 degrees) is expressed by:

$$h = \frac{KV^2}{2g} \text{ (in.)}$$

The head loss coefficient (K) for fittings having bends of 90 and 180 degrees commonly used in industry are K = 0.4 to 0.9 for 90 degree bends and K = 1.6 to 2.2 for 180 degree bends.

The flow path through the output ducts of the metering valves evaluated in this program were sharp 90-degree turns. Therefore the head loss using the above equation will only be a rough approximation and will be less than the actual losses resulting from testing. The smooth continuous radius ducts in the omega flex duct should give a much closer approximation of the calculated loss with test results.

2.2.3 Mechanical Properties

The mechanical properties which impact the metering valve design are:

a. Shape of the flexure channel

b. Size of the flexure channel (length, width and height)

c. Angle of deflection

d. Stress limits

e. Flexure spring rate requirement and spring natural frequency

f. Material selection

2.2.3.1 Shape - The four shapes of flexure-type metering valves were designed and tested. Two were flexible duct valves:

a. The omega shape, Figure 3(b)

b. The double gap or U-shape, Figure 3(c)

and two were flexure-type throttle valves:

a. The modified omega shape

b. The straight flexure-type, Figure 4

A feature which influenced the shape of each of these valves was that they were designed to hold the dimension of the metering gap constant with thermal growth of the flexure. Thermal growth of the flexure arm is compensated for by thermal growth of the side arms in the opposite direction, thus holding the metering gap dimension constant.

2.2.3.2 Size - The flex-duct valve is sized by the constraints of fuel flow requirements and metering response rate. The flow channel design constraints were presented in the flow calculations section. The metering response rate is a function of the flow channel length, cross-sectional area, channel wall thickness and material selection. A materials selection analysis is presented later in this section. The mechanical properties constraints which control the size of the valve are:

a. The angle of deflection required to meter fuel as required by the performance objectives.

b. The stress limits of the materials used in the flexure.

c. The flexure spring rate required to provide the necessary engine fuel control performance response.

These constraints are far more imposing on the sizing of a flex-duct channel than on a throttle valve flexure arm because the flexure section must be sufficiently large and strong to sustain high flow and pressure required by the ramjet engine.

2.2.3.3 Angle of Deflection - The angle of deflection of the flex-duct or flexure-type metering valve is best determined from testing. The objective is to determine the range of deflection of the metering valve flexure over which a linear or nearly linear flow relationship with valve deflection will be produced.

Typically, flexure-type valves have a range of flow linearity which is 60 to 75 percent of the total valve displacement as shown in Figure 11. The range of linearity is a function of valve design. The gap configuration at the metering edge is the controlling factor in determining the range of flow linearity.

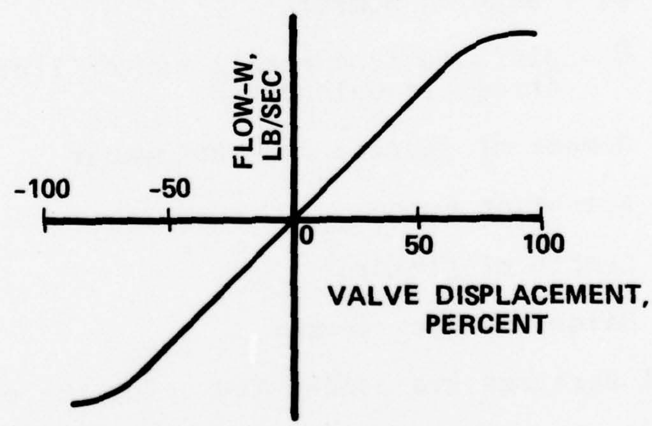


Figure 11. Flow linearity relation as a function of valve displacement.

2.2.3.4 Stress Calculations - Stresses occur in flexure-type throttle and flex-duct bypass valves from:

a. Bending of the flexure by normal modulation actuation. Bending resulting from internal pressure in the flow duct.

b. Torsion or twisting moment of the flexure, resulting from an offset of the actuation force. Torsion in the side flow channel or side flexure arms of the valve, which result from the normal modulation actuation.

c. Peeling of the braze connection in the flex-duct, resulting from pressure in the duct causing shear and bending stresses in the braze section between the laminated plates from which the flex-duct is constructed.

Bending stresses are calculated using the equation:

$$\sigma = \frac{Mc}{I}$$

where:

$M = F\ell$ = Bending moment

$c = \frac{h}{2}$ = Distance from member center line for which stress is calculated

I = Moment of inertia for the member

F = Actuation force

ℓ = Length of flexure

h = Height of flex member

Torsional stresses are calculated using the equation:

$$\tau = \frac{T}{2bht} \text{ for a hollow flex member}$$

$$\tau = \frac{T}{\sigma bh^2} \text{ for a solid flex member}$$

where:

$T = Fx$ = Twisting moment

b = Width of flex member

t = Wall thickness of hollow flex member

x = Offset distance of actuation force from the point in the plane of actuation force

The shear and bending stresses which result from the fluid pressure in a laminated flow duct occur along the interface braze line of the top and bottom laminates with the side walls as shown in Figure 12. The limit for the combined stresses causing peeling are approximated from experience at one tenth of the yield strength of the laminate material. A good design parameter is to select a side wall flow channel thickness which is three times that of the top and bottom thickness.

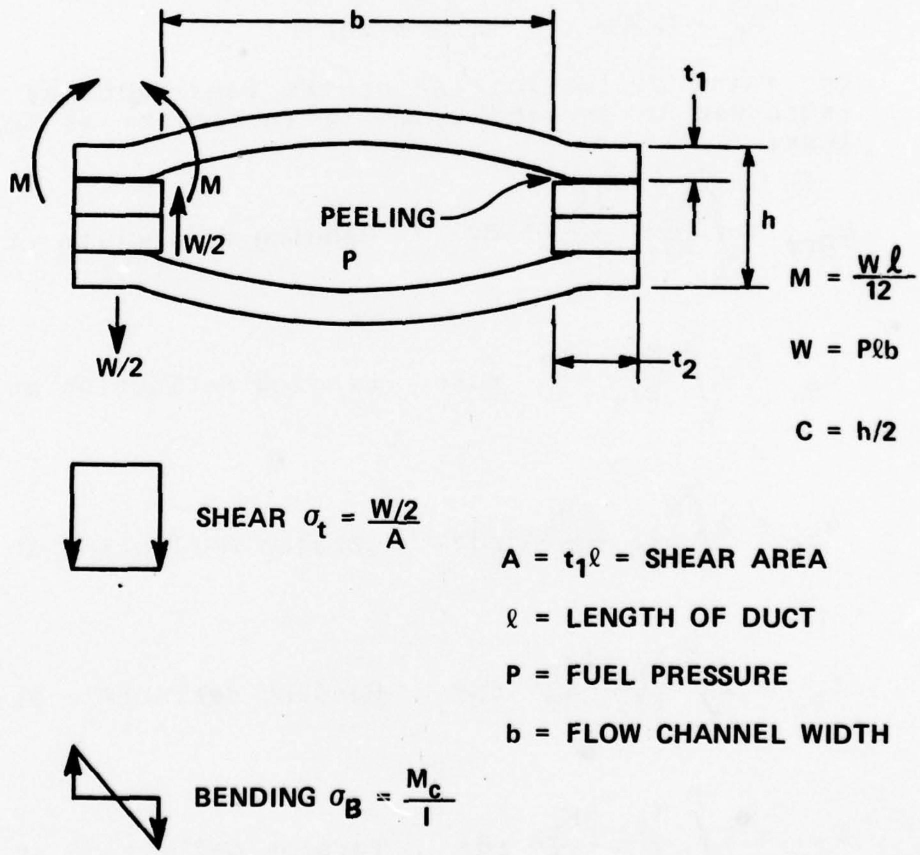


Figure 12. Shear and bending stress diagram resulting from peeling of laminates at braze.

The total stress in the area of braze between laminates is a sum of the shear and bending stresses.

$$\sigma_{TOTAL} = \sigma_t + \sigma_B = \frac{P\ell b}{2A} + \frac{P\ell^2 bh}{12I}$$

Flexure stress calculations were made for each of the flexure valves evaluated. The generalized stress calculations for each are expressed without the specific valve dimensions; therefore, they may be adjusted as desired for the sizing of a flexure valve for similar type application.

- o <u>Omega Valve Calculations</u> - The flex-duct dimensions, point of actuation force application, and areas of stress definition are shown in Figure 13. The bending and torsional moments which create the stress in the flexure and result in the deflection of the metering section are expressed as:

$$M = \frac{F}{2}(a + r \cos\theta)$$

$$M_B = M \sin\theta \qquad M_t = M \cos\theta$$

The total deflection (δ) of the flex-duct may be expressed in integral form in increments as follows:

$$\delta_{Arm} = \int_A^B \frac{M_{a+r}}{EI_1} \frac{\partial M_{a+r}}{\partial F} dy \qquad \text{Bending deflection at A}$$

$$\delta_{B_1} = 2\int_B^C \frac{M_{B_1}}{EI_2} \frac{\partial M_{B_1}}{\partial F} r d\theta \qquad \text{Bending deflection at B}$$

$$\delta_{t_1} = 2\int_B^C \frac{M_{t_1}}{J_2 G} \frac{\partial M_{t_1}}{\partial F} r d\theta \qquad \text{Torsion deflection at B}$$

$$\delta_{B_2} = 2\int_C^D \frac{M_{B_2}}{EI_2} \frac{\partial M_{B_2}}{\partial F} r d\theta \qquad \text{Bending deflection at C}$$

$$\delta_{t_2} = 2\int_C^D \frac{M_{t_2}}{J_2 G} \frac{\partial M_{t_2}}{\partial F} r d\theta \qquad \text{Torsion deflection at C}$$

where:

$$\frac{\partial M_{a+r}}{\partial F} = y$$

$$\frac{\partial M_{B_1}}{\partial F} = \frac{1}{2}(a + r\cos\theta)\sin\theta$$

$$\frac{\partial M_{t_1}}{\partial F} = \frac{1}{2}(a + r\cos\theta)\cos\theta$$

$$\frac{\partial M_{B_2}}{\partial F} = -\frac{1}{2}(r\cos\theta + a)\sin\theta$$

$$\frac{\partial M_{t_2}}{\partial F} = \frac{1}{2}(r\cos\theta + a)\cos\theta$$

These equations combined produce a total deflection at A, resulting from the application of a force F which may be expressed as:

$$\delta = \int_A^B \frac{Fy^2}{EI_1}\,dy + \frac{1}{2}\int_B^C \frac{Fr}{EI_2}\,(a + r\cos\theta)^2\sin^2\theta\,d\theta$$

$$+ \frac{1}{2}\int_B^C \frac{Fr}{J_2G}\,(a + r\cos\theta)^2\cos^2\theta\,d\theta$$

$$- \frac{1}{2}\int_C^D \frac{Fr}{EI_2}\,(r\cos\theta + a)^2\sin^2\theta\,d\theta$$

$$+ \frac{1}{2}\int_C^D \frac{Fr}{J_2G}\,(r\cos\theta + a)\cos^2\theta\,d\theta$$

The spring rate (K) of the flexure can then be expressed in terms of the actuation force (F) and the deflection (δ).

$$K = \frac{F}{\delta} = \frac{1}{\frac{1}{K_A} + \frac{1}{K_{B_1}} + \frac{1}{K_{t_1}} + \frac{1}{K_{B_2}} + \frac{1}{K_{t_2}}}$$

The bending stress at section 1 of this valve shown in Figure 13 is:

$$\sigma_1 = \frac{M_1 C_1}{I_1}$$

where:

$$C_1 = \frac{h}{2}$$

$$M_1 = (a + r)F$$

$$F = K\delta$$

At section two of Figure 13, the bending stress is:

$$\sigma_2 = \frac{M_2 C_2}{I_2}$$

$$M_2 = a\,\frac{F}{2}$$

$$C_2 = \frac{h}{2}$$

At section three of Figure 13 the torsional stress is:

$$\tau_3 = \frac{T}{2bh}$$

$$T = \frac{F}{2}\,(a + r)$$

At section four of Figure 13, the bending stress is:

$$\sigma_4 = \frac{M_B\,C_4}{I_2}$$

$$M_B = \frac{F}{2}\,(a + r\cos\theta)\sin\theta$$

$$C_4 = \frac{h}{2}$$

M - MOMENT

M_B - BENDING

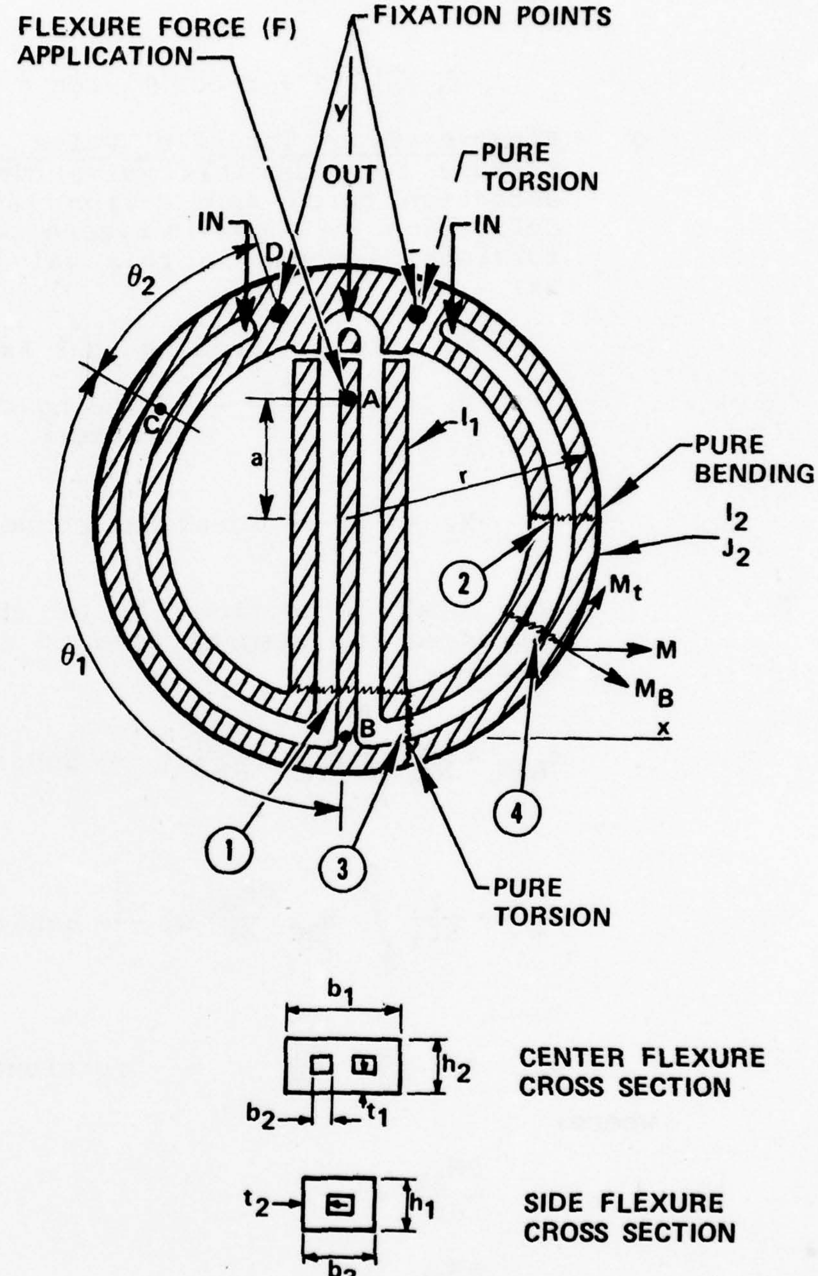
M_t - TORSION

I - INERTIA

J - POLAR MOMENT OF INERTIA

G - TORSIONAL MODULUS

E - MODULUS OF ELASTICITY



K_A - SPRING RATE IN FLEXURE AB

K_{B_1} - SPRING RATE FROM BENDING BC

K_{t_1} - SPRING RATE FROM TENSION BC

K_{B_2} - SPRING RATE FROM BENDING CD

K_{t_2} - SPRING RATE FROM TORSION CD

$$I_1 = \frac{1}{12}\left[b_1 h_2^3 - 2b_2 (h_2 - 2t_1)^3\right]$$

$$I_2 = \frac{1}{12}\left[b_3 h_1^3 - (b_3 - 2t_2)(h_1 - 2t_1)^3\right]$$

$$J_2 = I_2 + \frac{1}{12}\left[h_1 b_3^3 - (h_1 - 2t_4)(b_3 - 2t_2)\right]$$

Figure 13. Omega valve dimensions and stress locations.

At section four of Figure 13, the torsion stress is:

$$\tau_4 = \frac{F}{2}(a + r \cos\theta) \cos\theta$$

o **<u>Flexure Type Throttle Valve Calculations</u>** - The flexure type throttle valve dimensions, point of actuation force application, and area of stress definition are shown in Figure 14. The bending and torsional moments in this valve may be expressed as:

$$M_B = Fy \text{ --- Bending in Y Axis}$$

$$M_B = \frac{F}{2}y - \frac{F\ell_1}{2} \text{ --- Bending component in side flexure}$$

$$M_t = \frac{Fc}{2} \text{ --- Torsional component in side flexure}$$

The total deflection (δ) of the flexure may be expressed in integral form in increments as follows:

$$\delta_{ARM} = \frac{1}{EI_1}\int_A^B M_{AB} \frac{\partial M_{AB}}{\partial F} dy \text{ --- Bending deflection at A}$$

$$\delta_{B_1} = \frac{2}{EI_2}\int_B^C M_{BC} \frac{\partial M_{BC}}{\partial F} dy \text{ --- Bending deflection at B}$$

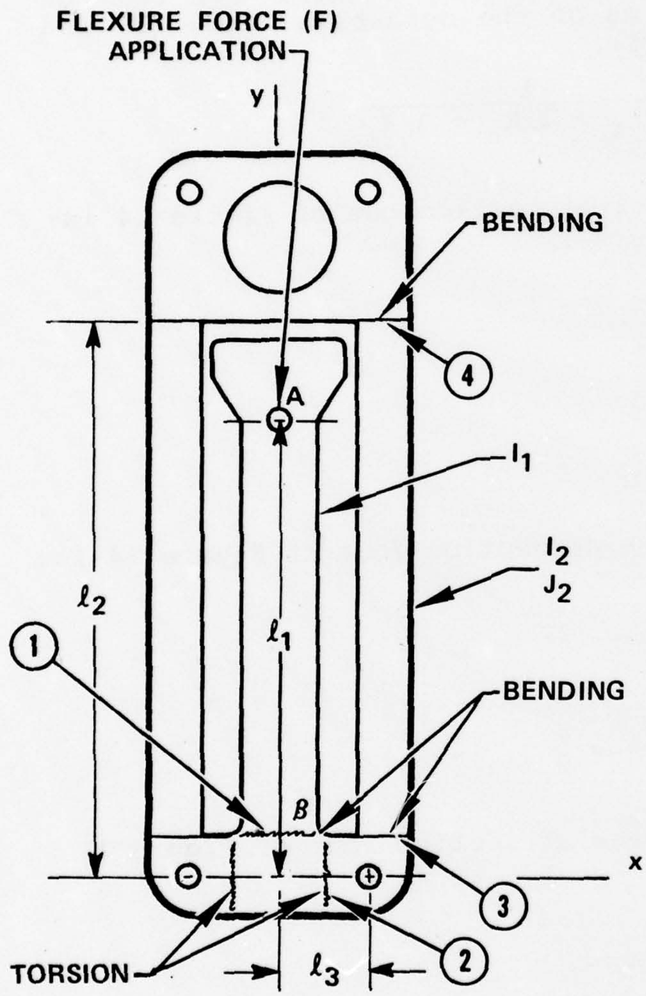
$$\delta_{t_1} = 2\,\frac{Tb}{GJ_2}\,\frac{\partial T}{\partial F} \text{ --- Torsional deflection at B}$$

where:

$$\frac{\partial M_{AB}}{\partial F} = y$$

$$\frac{\partial M_{BC}}{\partial F} = \frac{1}{2}(y - \ell_1)$$

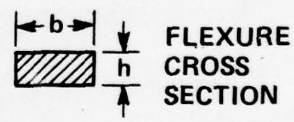
$$\frac{\partial T}{\partial F} = \frac{C}{2}$$



K_A - SPRING RATE IN AB

K_B - SPRING RATE FROM BENDING BC

K_t - SPRING RATE FROM TORSION BC



$c = \frac{h}{2}$

G - TORSIONAL MODULUS

E - MODULUS OF ELASTICITY

I - INERTIA

M_B - BENDING MOMENT

M_t - TORSION MOMENT

Figure 14. Flexure-type throttle valve dimensions and stress locations.

The equations combine to produce a total deflection at A resulting from the application of a force (F) which may be expressed as:

$$\delta = \frac{1}{EI_1} \int_A^B Fy^2 dy + \frac{1}{2EI_2} \int_B^C F\,(y - \ell_1)^2\, dy + \frac{Tbc}{GJ_2}$$

The spring rate (K) of the flexure can then be expressed in terms of the actuation force (F) and the deflection (δ).

$$K = \frac{F}{\delta} = \frac{1}{1/K_A + 1/K_B + 1/K_t}$$

The bending stress at section one of Figure 14 is:

$$\sigma_1 = \frac{M_{AB}c}{I}$$

where

$$c = \frac{h}{2}$$

$$M_{AB} = F\ell$$

The bending stress at section four of Figure 14 is:

$$\sigma_1 = \frac{M_c}{I}$$

where:

$$M = \frac{F\ell_2}{2} - \frac{F\ell_1}{2} = \frac{F}{2}(\ell_2 - \ell_1)$$

The torsional stress at section four of Figure 14 is:

$$\tau = \frac{T}{2bh}$$

where:

$$T = Fc/2$$

Bending and torsional stresses occur at section four; however, these stresses are small. The holes at the bottom of the flexure are guide holes and not mounting holes. The holes at the top of the flexure are mounting holes; therefore, as a force is applied at Point A on Figure 14, the flexure will move a small amount on the guide rods and flex at section four while being restrained at the mount holes.

- Double-Gap Valve Calculations - The stress calculations may be made from the spring rate calculations of the side arm flexure in much the same manner as in the case of the omega valve. The spring rate of the side flexures can be very well controlled; however, the high inertia in the flow body which results from the heavy housing for the flow channel produces modulation response rate that is very low. This heavy housing also results in high stress in the side flexure during flexure actuation. The proportional size of the side flexure and the flow channel housing is shown in Figure 15.

2.2.3.5 Materials Selection - Materials are selected for flex-type valves which consider the following qualities:

- Modulus of elasticity
- Torsional modulus
- Yield strength

The materials considered in this evaluation and the qualities of each are shown in Table 3.

TABLE 3. MATERIALS AND QUALITY EVALUATION

Material	E	G	Yield Strength
Stainless Steels	ksi x 10^3	ksi x 10^3	ksi
300 series	28.8	11.5	185
400 series	29.0	17.0	190
17-7	29.0	11.9	210

The material selected for the flexure laminates in this evaluation is 420 stainless steel. The yield strength is high with good modulus of elasticity and torsional modulus. A heat treat after braze given to the flexure type throttle valve flexure arm increases the ductility to improve its characteristics for cycle fatigue. Some loss of yield stress will be experienced from the heat treat; however, if the maximum actuation force level is low, the margin of stress below the yield will be more than acceptable for infinite life.

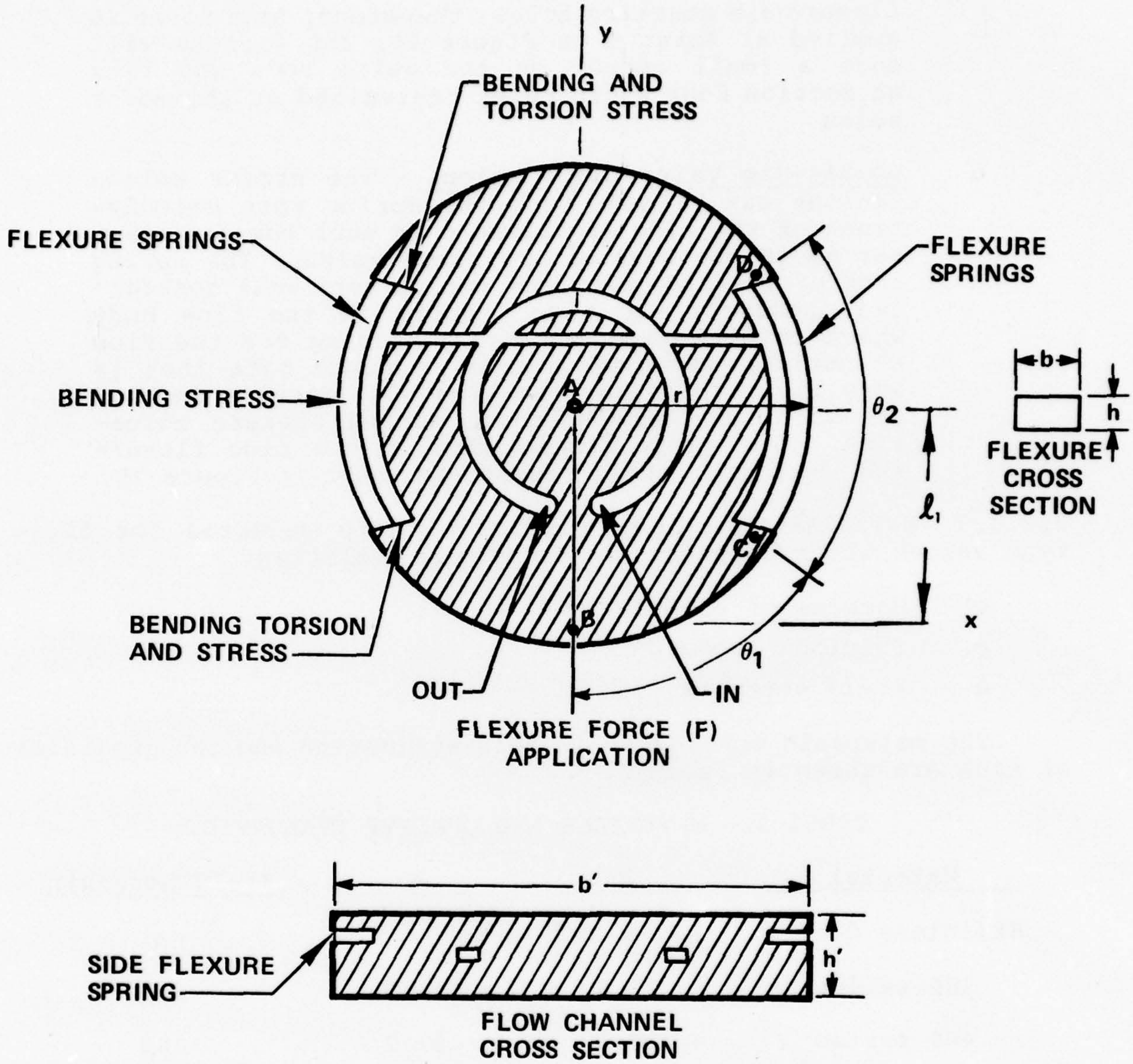


Figure 15. Double-gap valve dimensional stress locations.

3. TEST RESULTS

3.1 METERING VALVE CONFIGURATIONS

Four types of fuel metering valves were designed, fabricated, and tested for this program. Two of these valves were flex-duct valves and two were flexure-type throttle valves. The two flex-duct valves were the omega valve and the double-gap valve shown in Figures 16 and 17. The two flexure-type throttle valves were the omega valve modified and the straight flexure type throttle valve (see Figures 18 and 19).

3.2 TEST INSTRUMENTATION DESIGN

The parameters measured to determine metering valve performance were:

a. Flexure beam or duct movement

b. Flexure actuation force required

c. Flow rate

d. Pressure differential across the metering section

Schematics of the instrumentation to make these measurements are shown in Figures 20 and 21.

3.3 TEST CONCLUSION

From the sequence of development and test, it was determined that the straight flexure-type throttle valve, with a straight gap, best satisfied the performance requirements of the ramjet engine.

At the outset of the program, the bypass flex-duct valve appeared to possess characteristics which would meet the requirements of this application. Thermal growth, thought to be a problem with the straight-through flex-duct was overcome by the omega-shaped valve in the first design and double-gap, U-shape design in the second design. Thermal growth in the flow channel in one direction was offset by growth in the opposite direction.

The omega valve flow channel dimensions, to meet fuel requirements (flow and pressure) of the engine, resulted in a stiffness in the flexure section which produced an unacceptable valve actuation force level. This problem led to the design of the double-gap, flex-duct valve. The flexure sections of this valve are outside of the flow channel. This permits control of flexure dimensions to obtain the desired spring rate (see Figure 17).

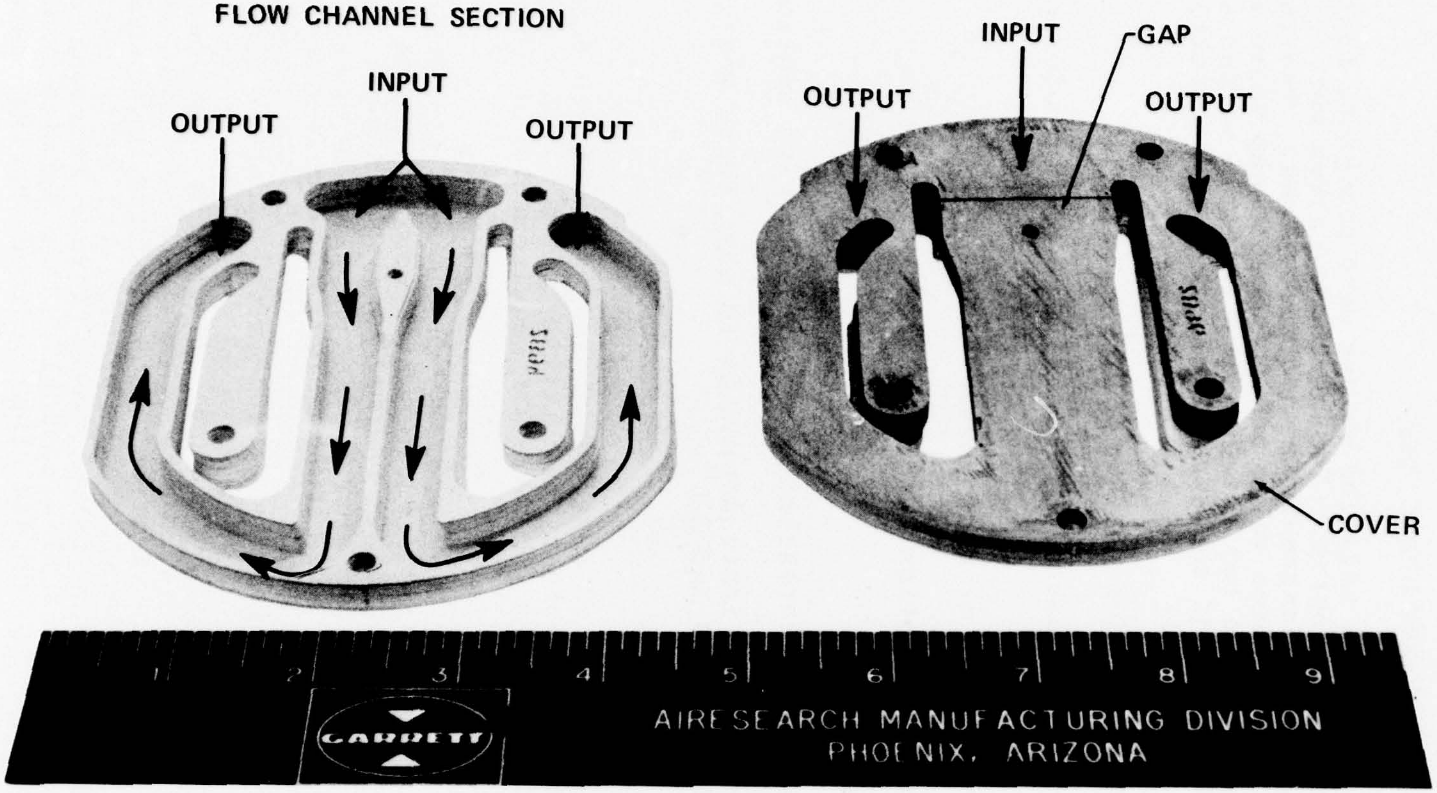


Figure 16. Omega flex-duct metering valve.

MP-66307

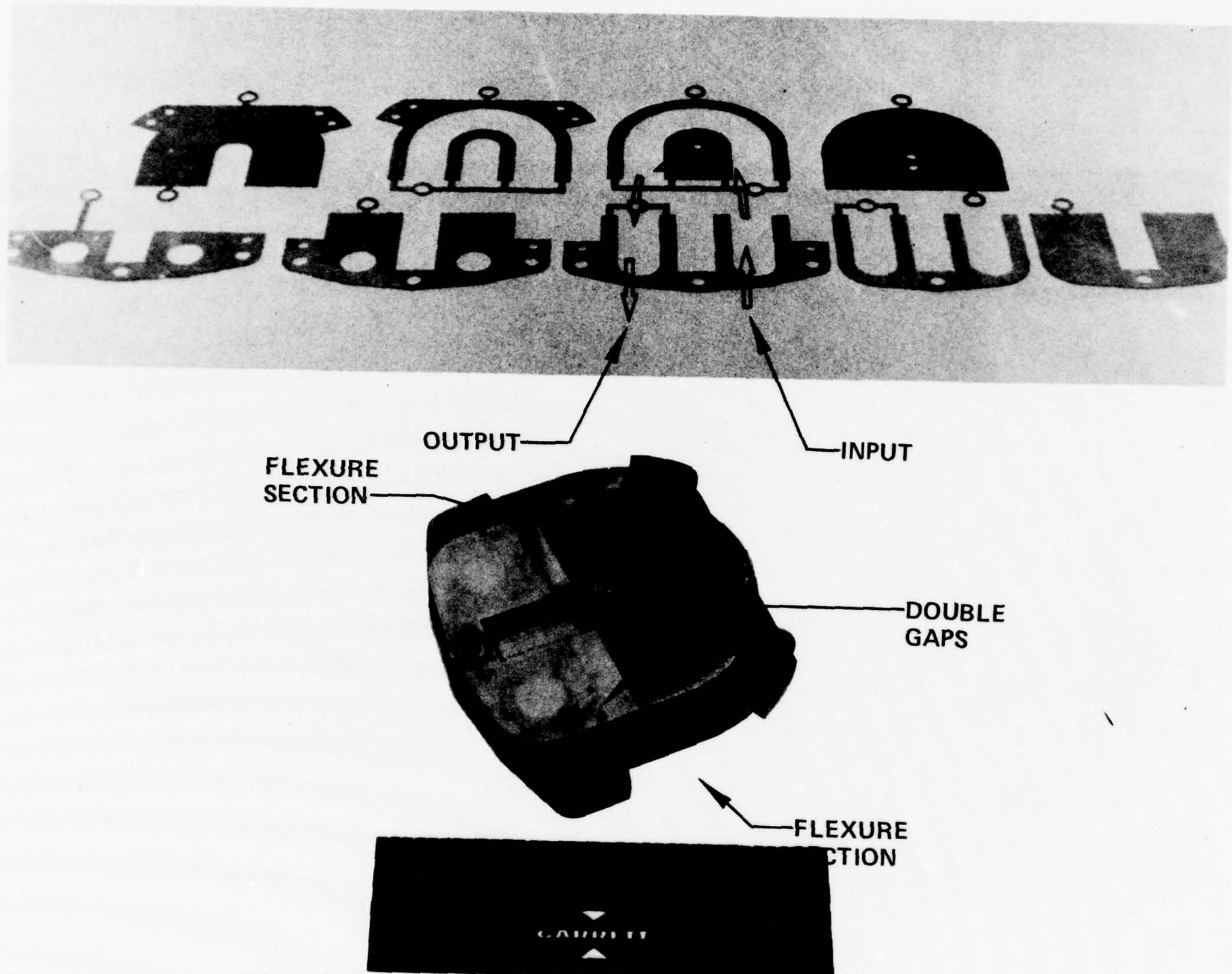


Figure 17. Double-gap flex-duct valve.

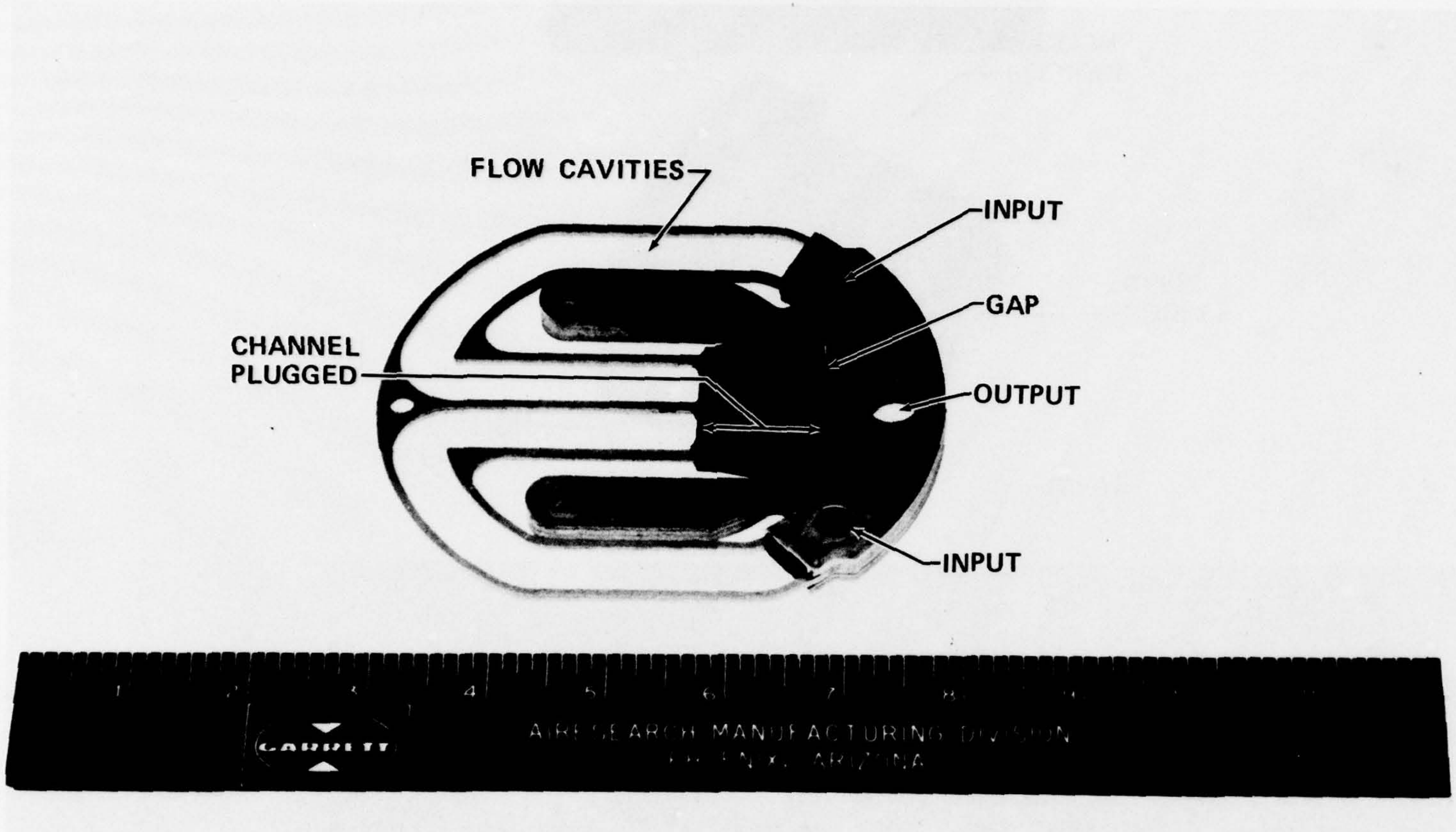


Figure 18. Flexure-type throttle valve (modified omega valve).

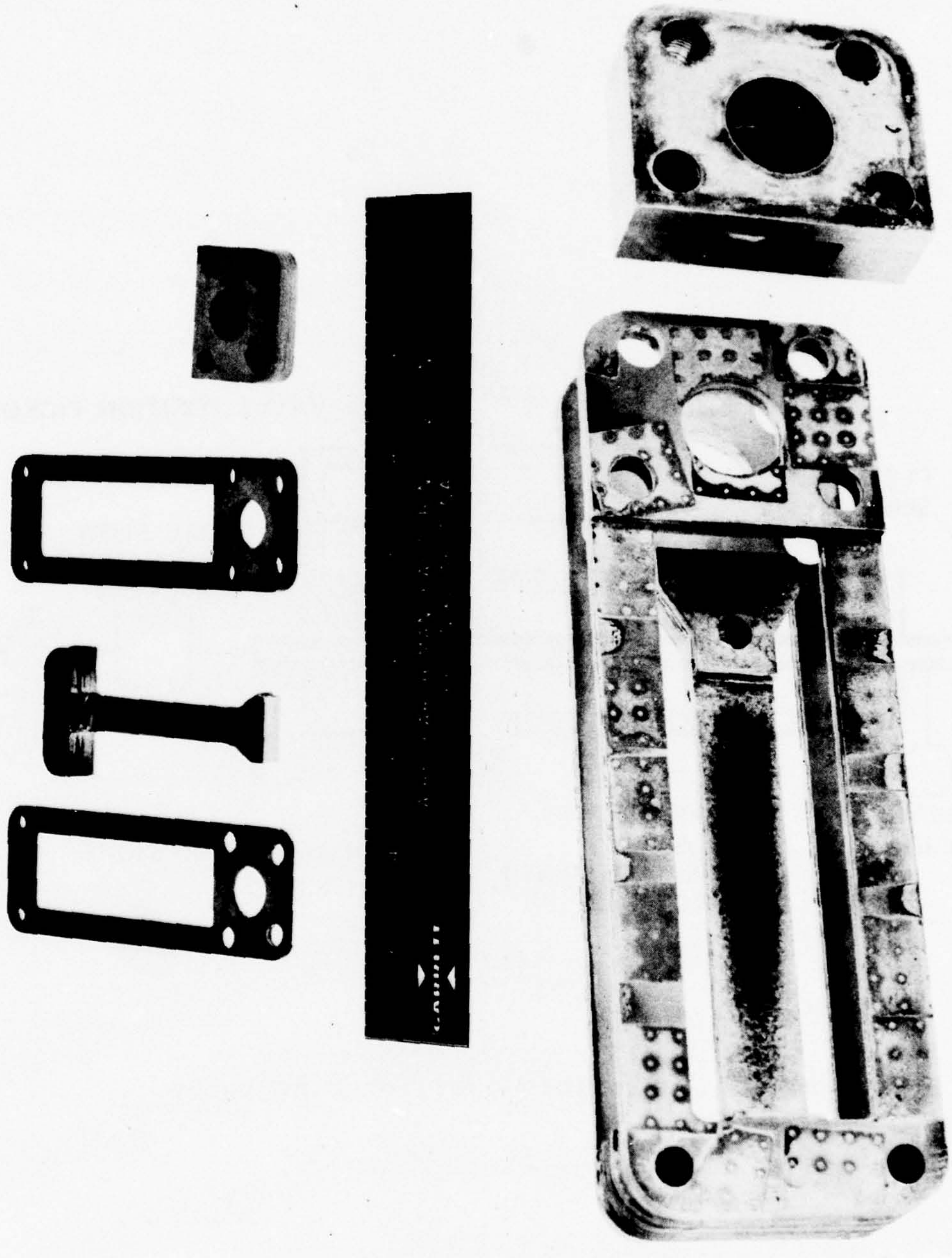


Figure 19. Straight flexure-type throttle valve.

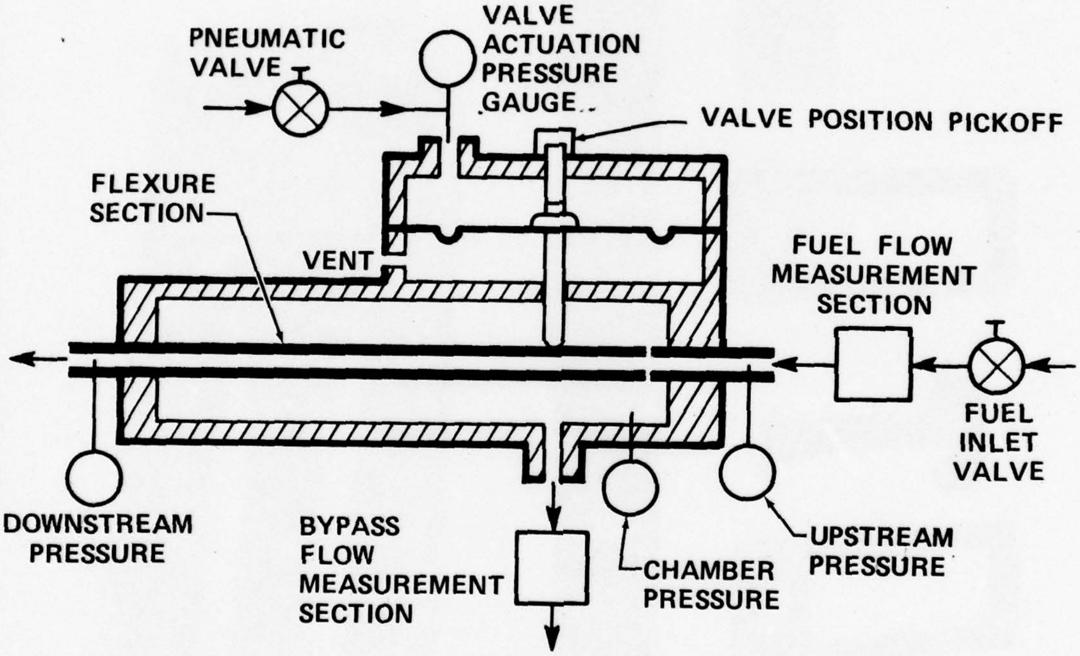


Figure 20. Flexible-duct metering valve.

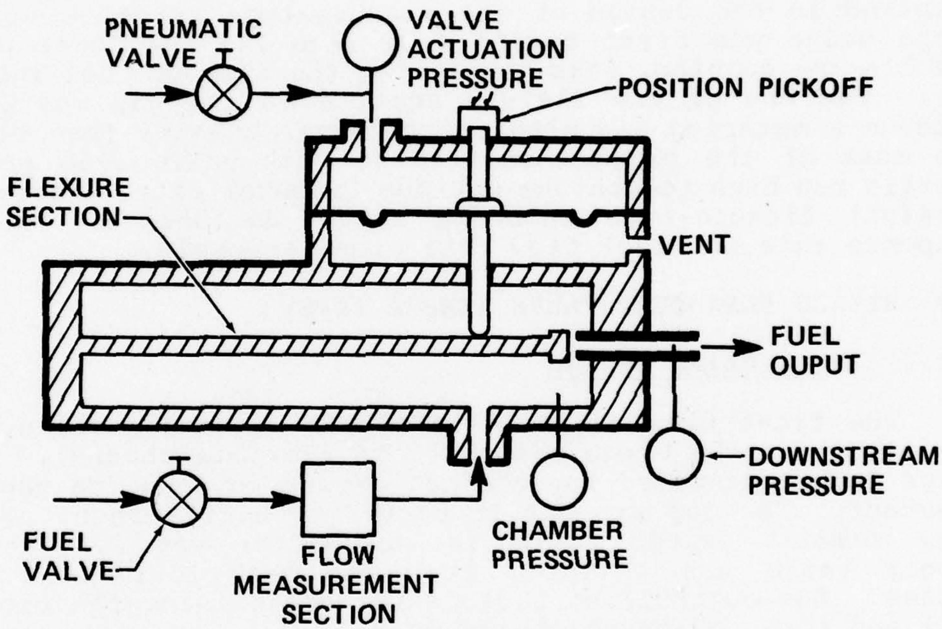


Figure 21. Flexure-type throttle valve.

The flow channel of this valve was made in two sections and designed to meet the flow requirements of the engine. The resultant inertia of the large mass in the moveable flow section lowered the valve response rate below the acceptable level. Another problem with this design was separation of the two flexure sections. This allowed an added and unsatisfactory degree of movement to the valve which complicated control of the metering area of the two metering gaps.

Test results of these two valves showed that the flow channel dimensions were the controlling factors in the valve design and resulted in high valve spring rate. This required a large actuation force or a high inertia which produced a response rate that was too low to meet engine fuel flow requirements. These findings resulted in the design of the flexure-type throttle valve. The omega valve was first modified by removing the lower portion of the flexure section, thus permitting the flow of fuel into a chamber. The end of the flexure section at the gap was plugged to produce a metering gap with the receiver channel (see Figure 18). The mass of the plugged section of this valve also produced an inertia too high to achieve optimum response rate. Therefore, the straight flexure-type throttle valve was designed to optimize response rate and fuel flow rate simultaneously.

3.4 BYPASS FLEX-DUCT VALVE (OMEGA TYPE)

3.4.1 Omega Valve Design

The first flex valve designed was the omega valve, deriving its name from the omega shape of the flexible channel. The omega shape (ω) compensated for thermal growth to keep the gap distance constant. The gap was cut to 0.0027 to 0.0029 inch. The design flow channel aspect ratio (height/width) was 0.4 inch. This aspect ratio was selected to minimize frictional fluid flow losses. The controlling factors for determining the flow channel wall and cover plate thickness were:

a. Maximum fuel pressure of 1500 psig (duct pressure)

b. Maximum flex-duct displacement of 75 percent of the channel height at the metering edge

c. Allowable bending stress limits and material modulus of elasticity

Two of these valves were built. The channel height of the first valve was 0.20 inch, resulting in a measured flexure spring rate of 1500 lb per in. The second was designed with a 0.14 inch channel height, with a resulting spring rate of 1000 lb per in. The spring rate of the second valve was significantly lower than the first; however, both were unacceptably high. The force level required to actuate these valves was too high and their size and weight were too large for the ramjet application.

The omega valve flow direction was determined from testing. The best direction of flow was determined to be from the stationary section across the metering gap and into the flexure section (see Figure 16). The reasons for this are:

a. The stationary section can better be designed for the higher fuel pressure upstream of the metering gap than the flexure section. This permits a lighter wall construction of the flexure channel, thus reducing its stiffness.

b. Tests show that the bypass fuel flows out of the lower flex duct section into the chamber over the flexure and into the receiving port at the top of the metering gap when the flow direction is through the flexure section and into the receiver channel. This significantly degrades the metering performance. The valve could be redesigned to prevent this recirculation; however, it was much simpler to flow fuel in the opposite direction.

3.4.2 Omega Valve Performance

The performance of the valve with the 0.20-inch channel height is shown in Figures 22 and 23. The flex duct was maintained in the zero-deflection position throughout these tests, and the fuel inlet pressure was varied so as to compare the actual performance with that predicted. These two graphs show that the actual results were substantially below the predicted performances. The cause for this is that the channel losses were greater than anticipated in the 180-degree bends. From Figure 23 it may be observed that the optimum fuel pressure ratio (P_{out}/P_{in}) to achieve maximum power efficiency is between 0.65 and 0.7.

The omega valve with the lower flexure spring rate (1000 lb per in.) was used to develop performance data during metering. The minimum actuation force required to displace the metering flexure to modulate fuel flow, as shown in Figure 24, was 133 lb. Significant hysteresis existed at high flow, but was very small at minimum flow. The flow difference between inlet and output flow was the bypass flow recirculated back to the pump. The force required to actuate the flex-duct was the same whether fuel was flowing through the valve or not.

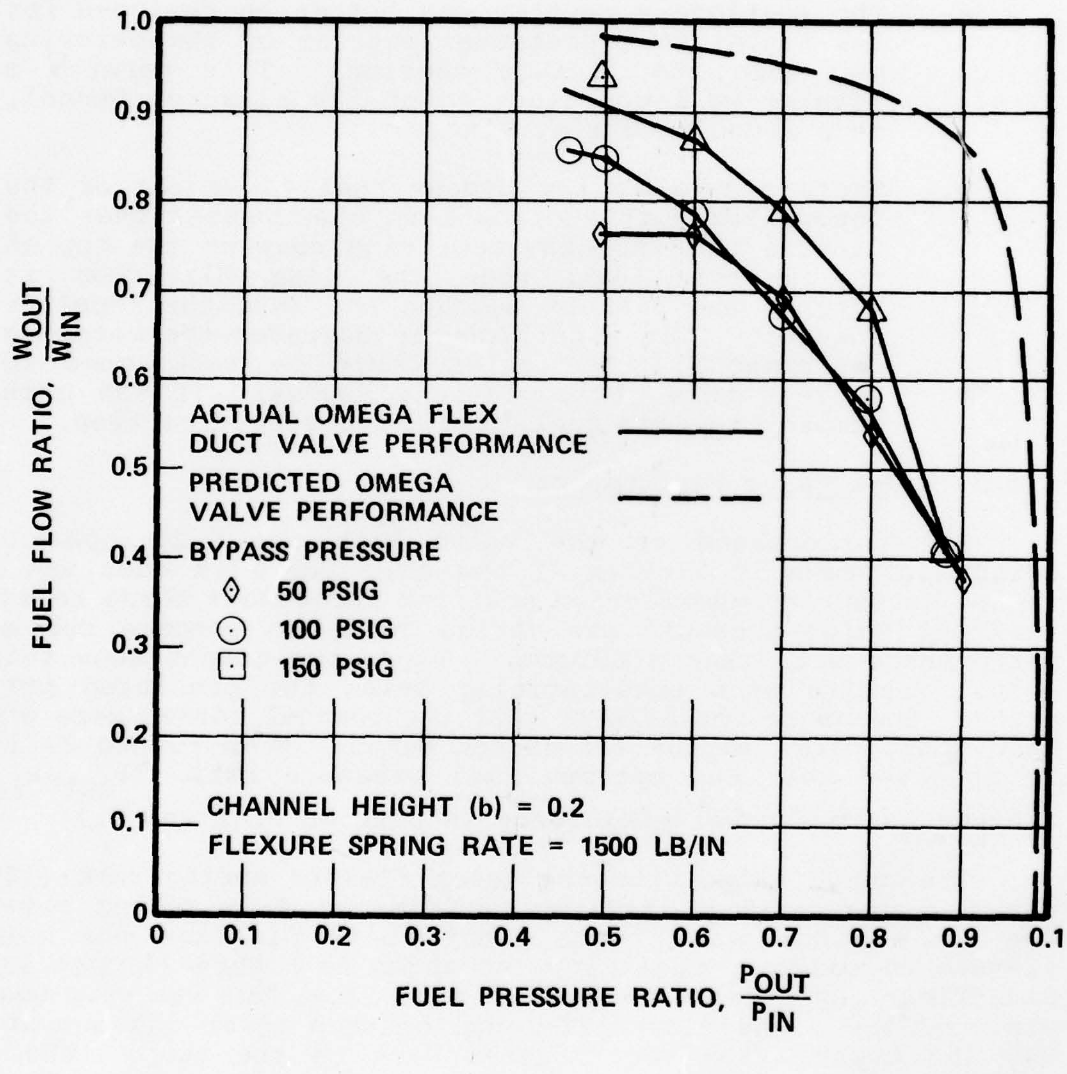


Figure 22. Omega-type flex-duct valve flow performance.

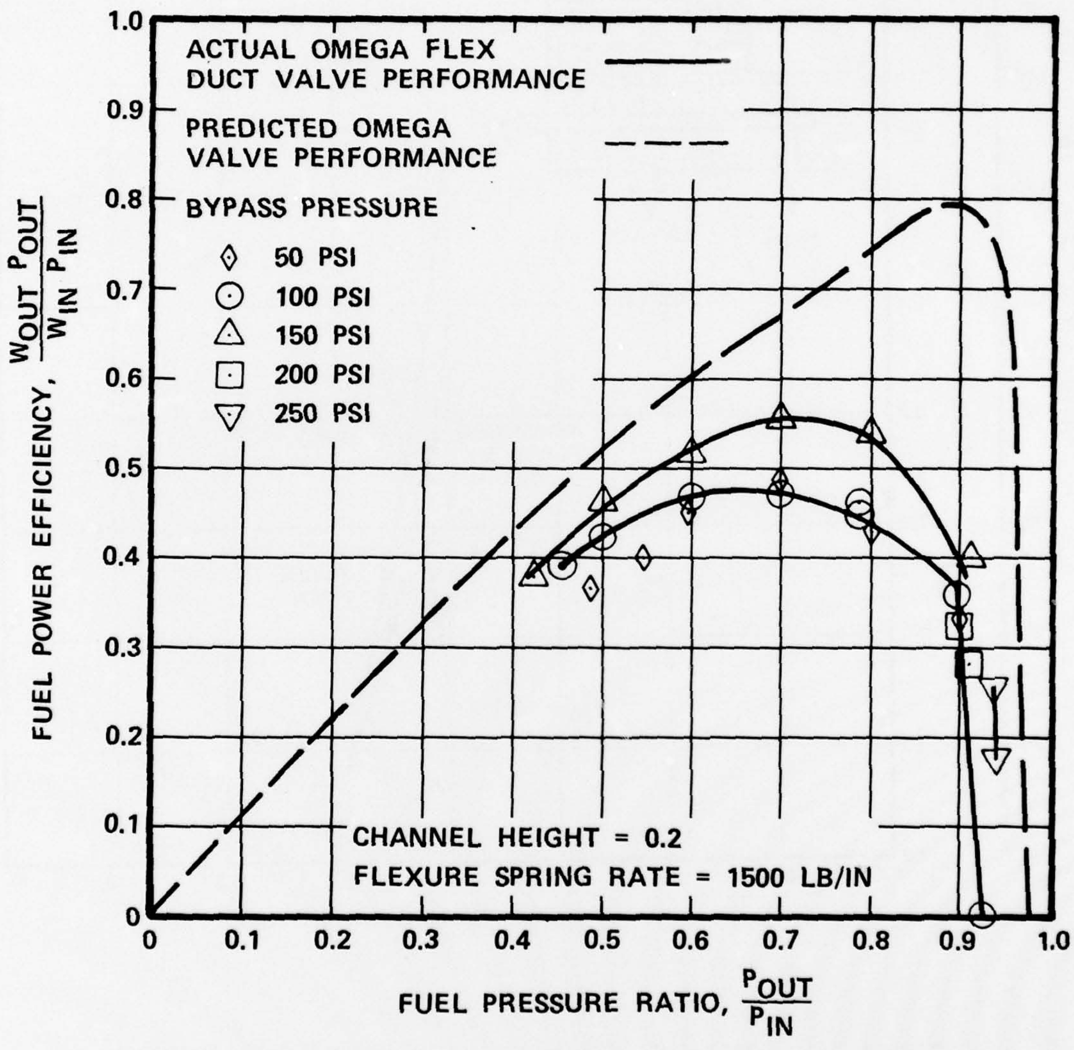


Figure 23. Omega-type flex-duct valve power efficiency.

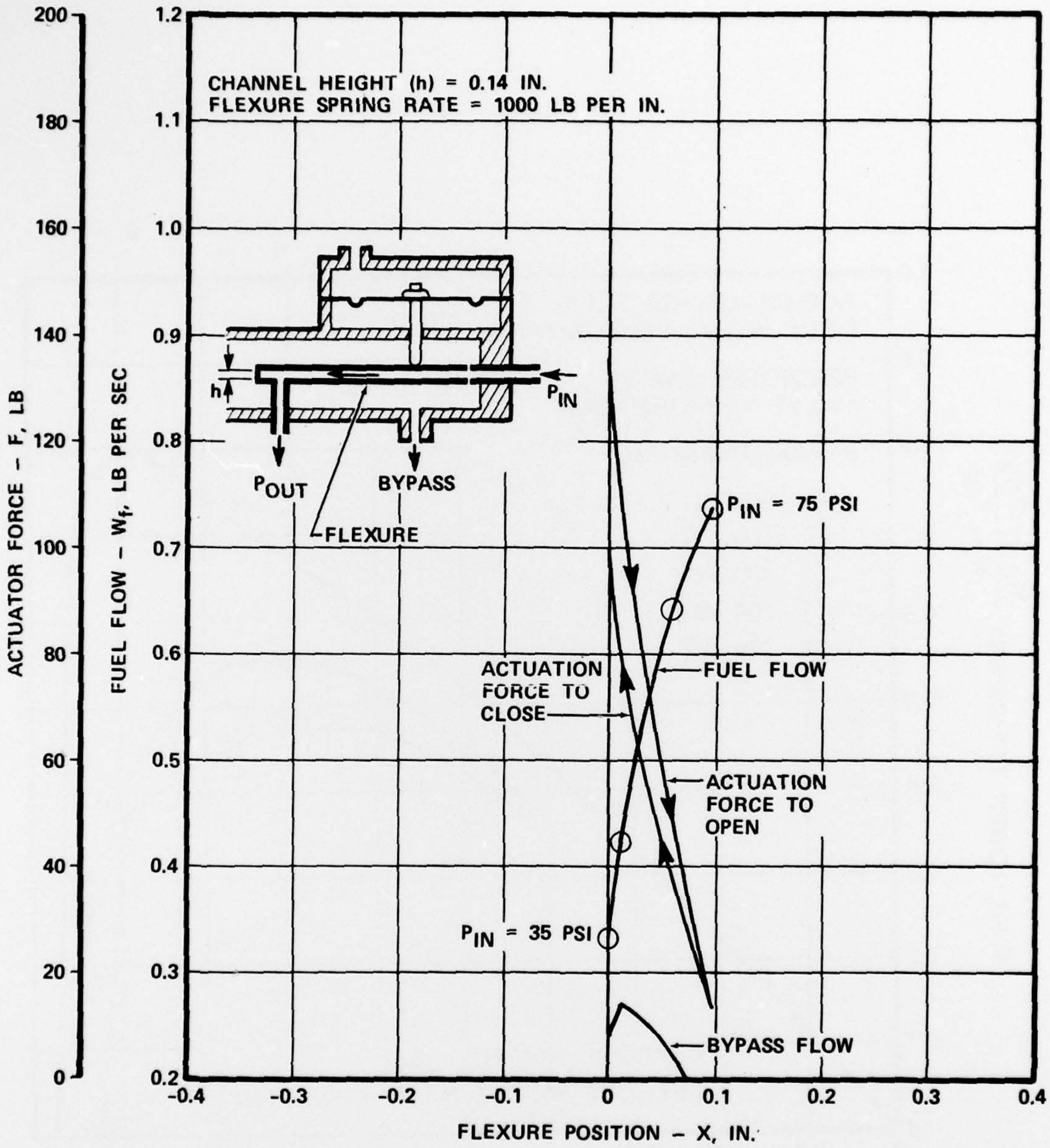


Figure 24. Omega-type flex-duct valve metering performance.

3.5 BYPASS FLEX-DUCT VALVE (DOUBLE-GAP TYPE)

3.5.1 Double-Gap Valve Design

The double-gap valve was designed primarily as an attempt to reduce the spring rate of the bypass valve and maintain the temperature compensation feature of the omega valve. The basic design is a flow channel separated from the flexure springs as shown in Figure 3(c). The U-shaped channel is fastened to the stationary assembly containing the input and output ports with small flexible arms. An additional spring is placed under the moveable section to position the valve at null.

The stress in the flow channel, unlike the omega valve, is a function of the internal pressure only because there is no bending in the flow channel.

The metering gap was cut and tested at 85, 90, and 95 degrees to study the effect on metering performance.

3.5.2 Double-Gap Valve Performance

A valve with the straight-cut gap (90 degree) was the first to be tested. This valve required a large gap to permit sufficient valve displacement to achieve minimum flow; therefore, excessive leakage flow in the null position resulted. The two angle-cut valves were then tested. The 95-degree gap was first tested with the flow channel of the flexure section aligned with the flow channel of the stationary section. A coil spring preload under the flexure section was required to close the gap and reduce leakage for maximum flow (see Figure 25).

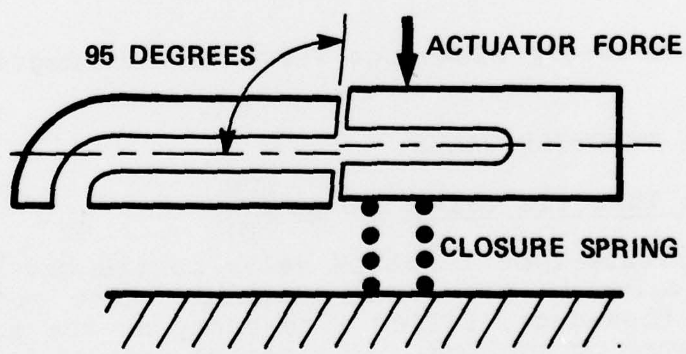


Figure 25. Double-gap flex-duct valve with 95-degree gap.

Sizing the closure spring with sufficient spring rate to close the gap to obtain maximum flow required an excessively high actuation force for modulation to minimum flow.

A double-gap valve with an 85-degree angle cut gap, shown in Figure 26, was fabricated with the flow channel of the flexure section offset from the flow channel of the stationary section. A restoring spring replaced the closure spring so that the actuation force required increased as the valve was opened to the full-flow position. This configuration was an improvement over the 95- and 90-degree gap valves; however, separation of the two leaf springs, one on each side of the valve, allowed a second degree of movement around the flow axis. This caused a differential and variable flow metering at each of the two gaps. Also, the natural frequency of the valve was 50 Hz which was less than desirable for the response rate required for the ramjet fuel control application. This latter deficiency led to the investigation of the flexure-type throttle valve which would permit better control of the flexure spring and produce a suitable response rate for the ramjet engine.

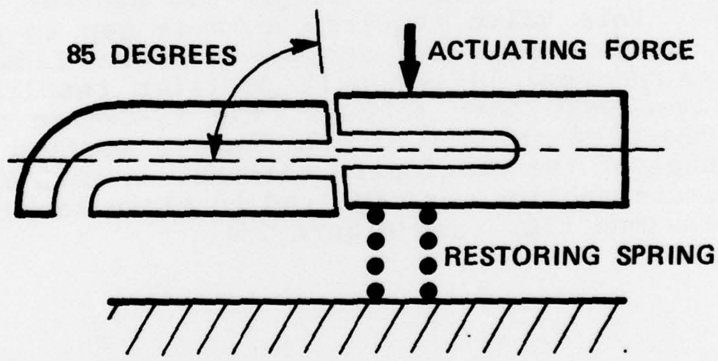


Figure 26. Double-gap flex-duct valve with 85-degree gap.

3.6 FLEXURE TYPE THROTTLE VALVE

3.6.1 Omega Type Throttle Valve Design

The first flexure-type throttle valve tested was a modified omega flex-duct valve to prove the throttle valve concept (see Figure 18). The flex-duct, filled with epoxy at the face of the gap to form the metering section, had a spring rate of 5 lb per in. The dimensions of the metering section receiver channel were the same as the flex-duct metering valve. A cross section of the flexure arm and metering throttle is shown in Figure 27.

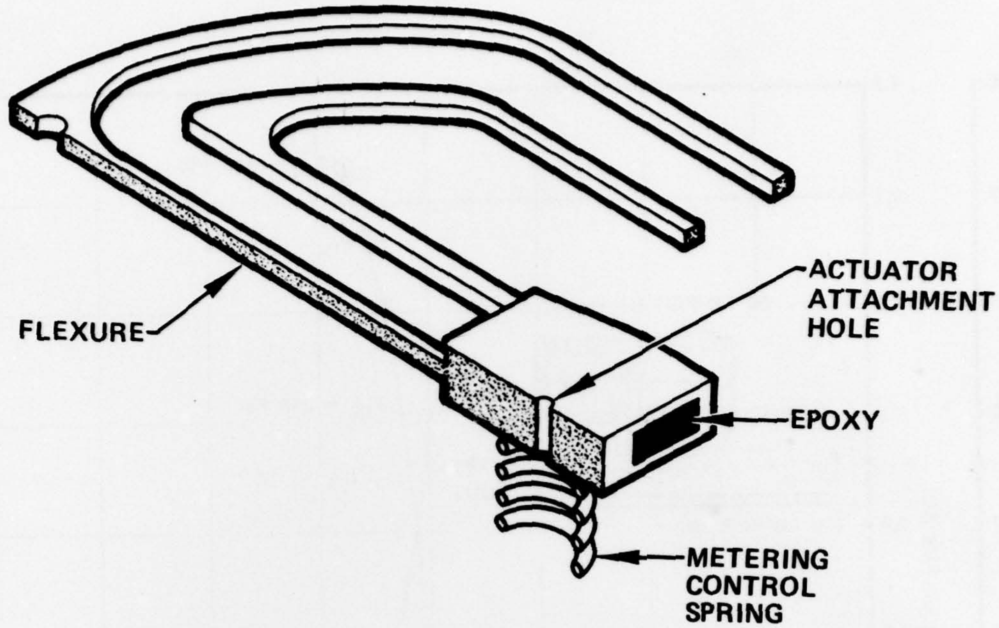


Figure 27. Cross section of the omega-type throttle valve.

Removing the lower flex-duct channel section substantially reduced the spring rate of the flexure section. Therefore, a coil-type metering control spring (see Figure 27) was added to provide a resistive force against which the valve actuator reacted to meter fuel.

3.6.2 Omega-Type Throttle Valve Performance

The performance shown in Figure 28 was obtained by pressurizing the valve chamber to 340 psi. The flexure was positioned to obtain minimum flow (closed) by exerting a force on the actuator pin to hold a preload against the lower spring as shown in the sketch on Figure 28. Flow was increased by reducing the actuator force. The flow characteristics shown are satisfactory; however, the heavy mass of the throttle section and thus high inertia resulted in a low frequency response rate of the valve. This testing served to show that the throttle-type metering valve could perform the fuel metering functions required for the ALVRJ fuel control. The valve however would require redesign to reduce the heavy mass at the throttle section to increase the valve response rate. To reduce the overall size of the throttle valve, a simple straight flexure-type throttle valve was designed (Figure 19). This valve easily permits achieving the desired response rate by sizing a coil spring in series with the flexure spring.

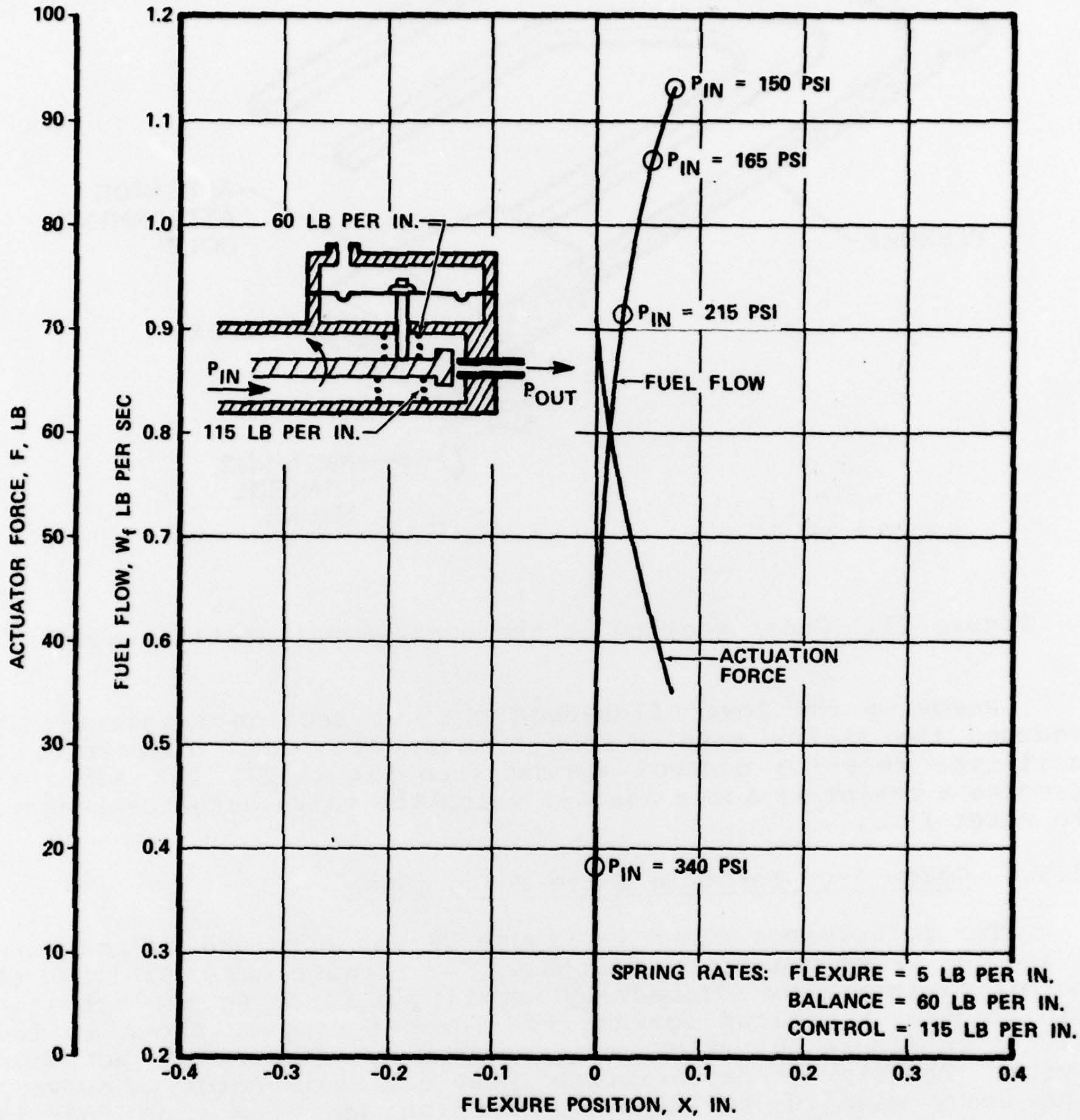


Figure 28. Omega-type throttle valve metering performance.

3.6.3 Straight Flexure-Type Throttle Valve Design

This compact valve was designed to make use of desirable characteristics learned from testing the various configurations of omega and double-gap flex-duct valves and the modified omega-type throttle valve. It was designed to achieve a desirably high response rate with minimum actuation force to obtain the flow rates specified. It is lightweight and low cost. The flexure W-shape was used for temperature compensation and low weight and to permit easy positioning of the metering edge relative to the stationary port. The measured center flexible arm spring rate (K_{CA}) was 56 lb per in. and the spring rate for the side positioning arms (K_{SA}) was 168 lb per in. Therefore, from the equation $K = \frac{1}{1/K_{CA} + 1/K_{SA}}$, the overall calculated spring rate was 42 lb per in. The positioning arms allow for increased stability to insure that the valve motion was in one direction only. These arms were designed such that the center flexible arm travels between the side arms, thus allowing for a more compact valve than the bypass valve (see Figure 29). The valve is a four-piece assembly (as seen from Figure 19), and the metering edge of the flexible arm and stationary port were surface ground for close tolerance control of the metering gap.

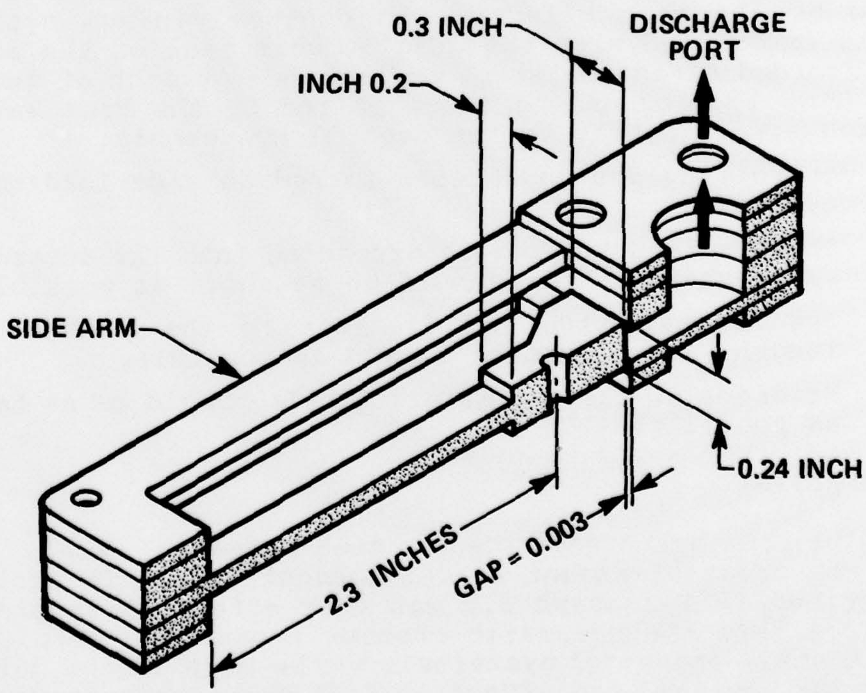


Figure 29. Straight compact flexure-type throttle valve cross section.

3.6.4 Straight Flexure-Type Throttle Valve Performance

Performance data for the straight flexure-type throttle valve using MIL-C-7024, Type II calibrating fluid is shown in Figure 30. This data was recorded using a fuel inlet pressure of 400 psi; however, with a variation in inlet pressure from 200 to 800 psi, the fuel flow remained constant at each of the five required performance operating points shown. The low value of flexure spring rate (42 lb per in.) as shown in Figure 31 allows for a selection of a control spring to achieve any valve response rate required to satisfy the fuel system requirements. It also permits the design of a servovalve with low levels of force for valve modulation (34 pounds maximum) as shown in Figure 31. The fuel in the chamber surrounding the flexure arm serves as a damper and adds stability to the metering valve. This metering valve design resulted in a simple, low cost valve which may be adapted to multiple applications for ramjet engine fuel controls.

3.7 FLEXURE ACTUATION ROD

The flexure actuation rod, which is designed to deflect the metering flexure, is actuated by pneumatic pressure behind a diaphragm. The hole in the housing of the valve through which the actuation rod slides was sealed by either a triangular or a U-shaped Teflon seal backed up by an O-ring. The triangular seal proved to be the most effective to achieve minimum hysteresis. Hysteresis associated with the drag of this seal on the actuation rod was recorded during metering performance of each of the valves tested. Side loading on the actuator rod by the flex valve also contributed significantly to the overall hysteresis.

The controlling design factors to reduce side loading on the actuator rod are:

a. The unguided rod length extending into the metering valve chamber. It should be as short as possible.

b. The ratio of the rod diameter to the guided rod length. It should be as low as possible.

c. Hardness of the rod material. It should be as hard as possible.

3.8 HYSTERESIS

Resolution hysteresis and high-pressure seal leakage required the greatest amount of development time. The triangular seal described in Paragraph 3.7 was very effective in preventing fuel leakage into the pneumatic chamber around the actuator diaphragm. A small amount of hysteresis maybe seen in the data shown in Figure 30; however, the effect on fuel metering performance was negligible. The displacement between the two data points of each altitude is the system hysteresis. The effect of hysteresis is slightly greater at low flow (high altitude) conditions.

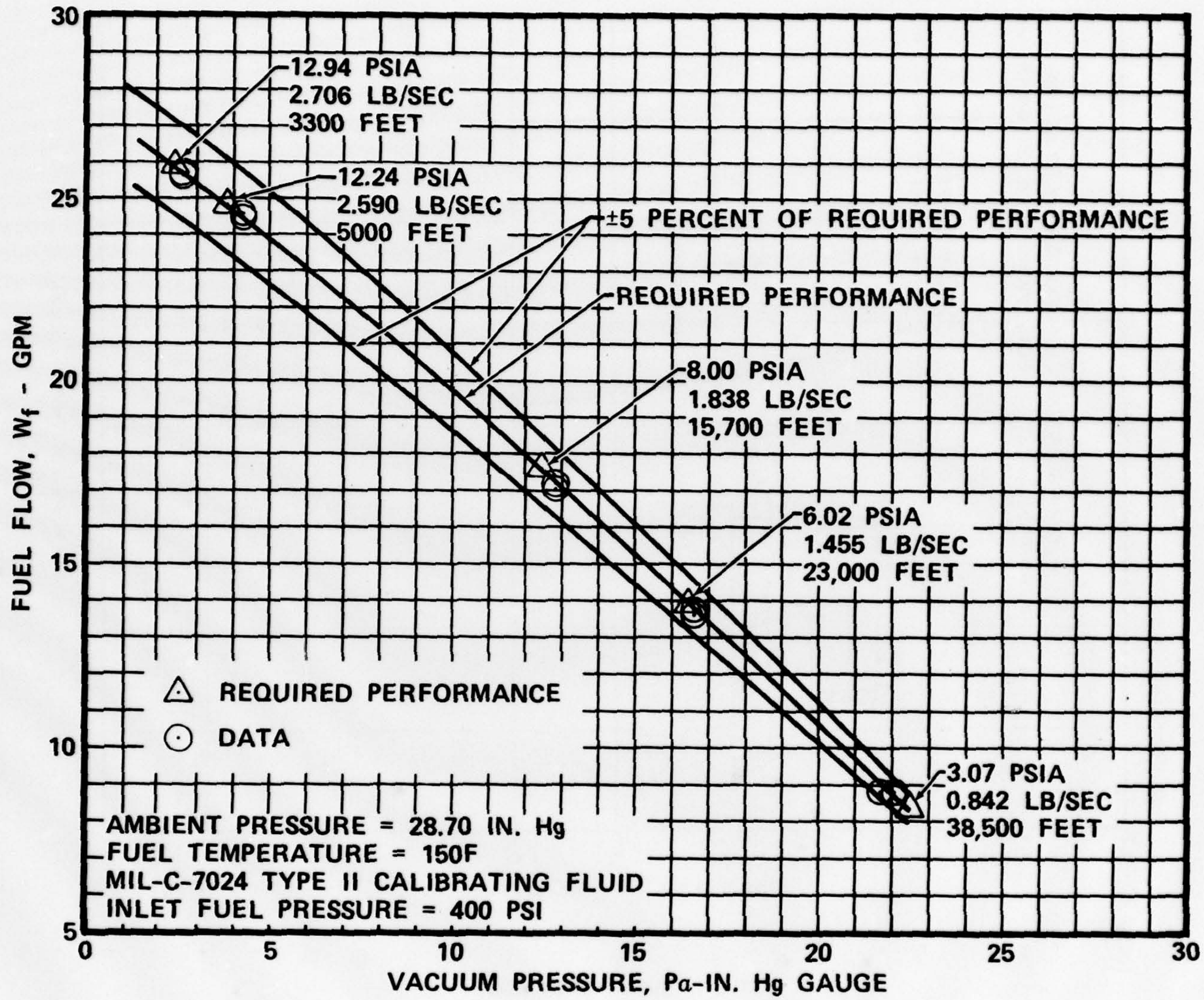


Figure 30. Straight flexure-type throttle valve performance.

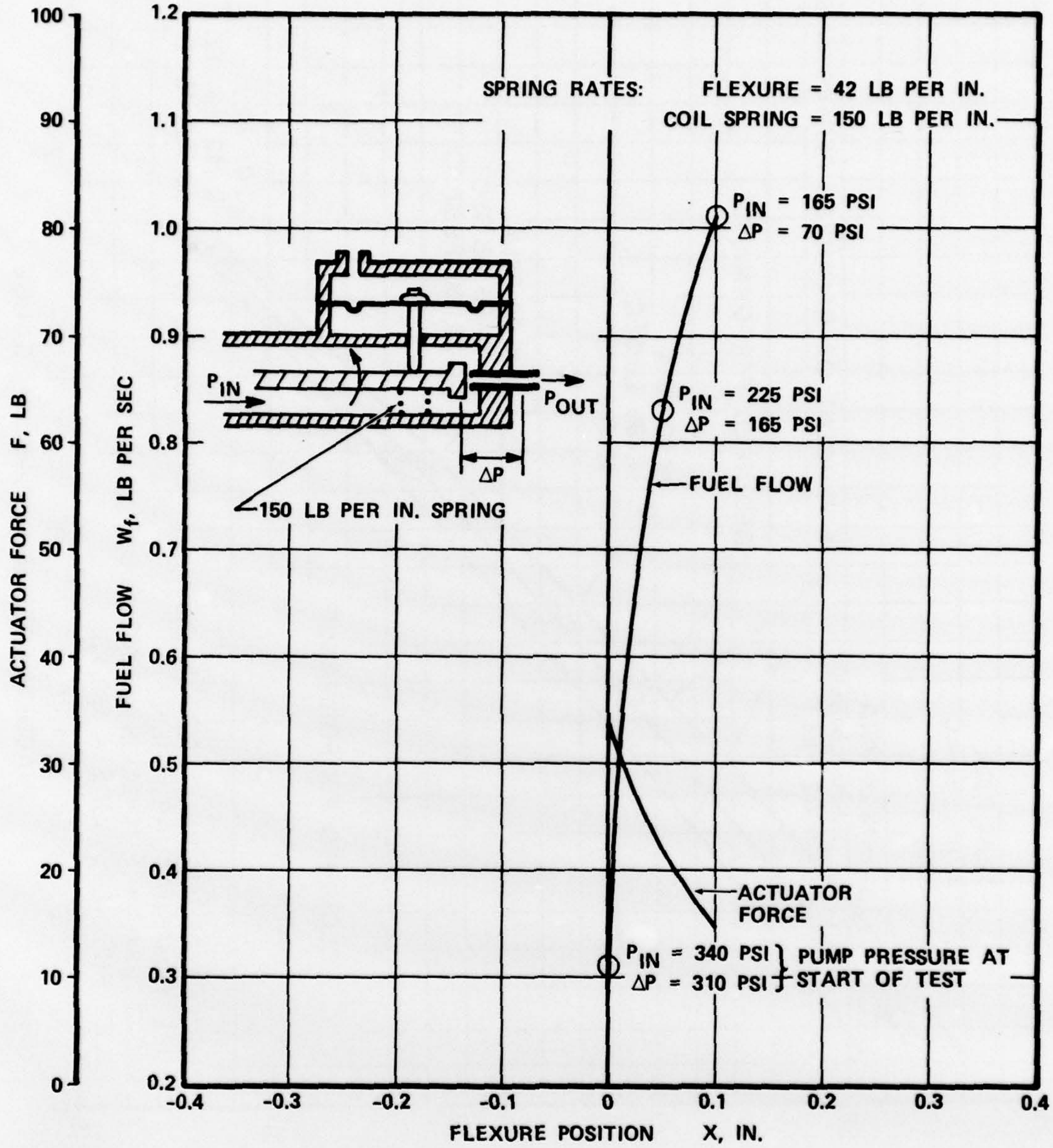


Figure 31. Straight flexure-type throttle valve actuator force and fuel flow versus flexure position.

4. CONCLUSIONS

The analytical evaluation of the flex-duct valve was successful in that tests confirmed the equations for the design of the valve. The main conclusion reached was that, for applications where large flows or high pressures are required, the hollow tube flex-duct valve is not suitable. Its use is primarily for low pressure, low flow, or servo applications. For high flow applications, the solid cantilever throttle valve is applicable, as shown by the test program on the units sized for the ramjet fuel metering application.

This valve, although small, lightweight, and simply constructed, is sufficiently sturdy to withstand a fuel pressure of 1500 psi. A turndown ratio of 3.2:1 (0.84 lb per sec minimum flow to 2.71 lb per sec maximum flow) is easily obtained with a straight cut at the metering gap. Higher turndown ratios can be achieved by forming the gap on an angle. This will permit significant reduction of the minimum flow and thereby increase the turndown ratio. Hysteresis in the flexure actuator is the controlling factor in determining the minimum fuel flow limit. The maximum turndown ratio achieved with a straight cut gap was 4.8. The minimum flow achieved was 0.57 lb per sec. Minimum fuel flow is limited by the design of the smallest metering gap which will still permit the maximum flow limit to be achieved.

The omega-type flex-duct valve is limited in its application as a fuel metering valve for ramjet engines because of the high spring rate which resulted from the large wall thickness required to withstand the high fuel pressure requirements. The valve is heavy and large and the actuation force to modulate the valve is high.

The double-gap metering valve is not recommended because it also is heavy and large. The double gap complicates control of the metering gap during modulation. The flexure leaf springs may be designed to achieve a low spring rate; however, the high inertia of the movable duct section more than offsets the advantage of the low spring rate in designing to achieve a high response rate system. The force levels to properly operate this valve are high and only slightly better than the omega type flex-duct valve.

The flexure actuation technique used in the testing of the valves in this program was satisfactory to achieve ±5 percent flow accuracy over the range of flow from 0.57 to 2.71 lb per sec. To further reduce the flow rate, however, an actuation technique should be designed to further reduce hysteresis. The hysteresis band of the throttle valve with this actuator was ±0.013 lb per sec.

5. RECOMMENDATIONS

Should a minimum fuel flow requirement exist below 0.84 lb per sec, it is recommended that the flexure-type throttle valve be studied to angle cut the metering gap between the throttling head and the receiver port. This will achieve an increase in the turndown ratio without increasing the maximum flow.

It is also recommended that the flexure actuator be designed to reduce hysteresis to ±0.006 lb per sec or less if a turndown ratio of 20:1 and a five percent fuel flow accuracy are required.

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